TRANSACTIONS

OF THE

AMERICAN SOCIETY

OF

MECHANICAL ENGINEERS.

VOL. XX.

XXXVIIITH MEETING, NEW YORK CITY, 1898.
XXXIXTH MEETING, WASHINGTON, D. C., 1899.

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OFFICERS

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

1898–1899,

FORMING THE STATUTORY COUNCIL.

PRESIDENT.

GEORGE W. MELVILLE.....Washington, D. C.

VICE-PRESIDENTS.

CHAS. M. JARVIS.....Berlin, Conn.
JOHN C. KAFER.....New York City.
WALTER S. RUSSEL.....Detroit, Mich.
Terms expire at Annual Meeting of 1899.

E. D. MEIER.....St. Louis, Mo.
GEORGE R. STETSON.....New Bedford, Mass.
B. H. WARREN.....Pittsburg, Pa.
Terms expire at Annual Meeting of 1900.

MANAGERS.

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GUS. C. HENNING.....New York City.
A. WELLS ROBINSON.....So. Milwaukee, Wis.
Terms expire at Annual Meeting of 1899.

GEO. RICHMOND.....New York City.
JAS. B. STANWOOD.....Cincinnati, Ohio.
HENRY H. SUPLEE.....New York City.
Terms expire at Annual Meeting of 1900.

EDGAR C. FELTON.....Harrisburgh, Pa.
A. M. GOODALE.....Waltham, Mass.
R. H. SOULE.....Chicago, Ill.
Terms expire at Annual Meeting of 1901.

TREASURER.

WM. H. WILEY.....No. 53 East 10th St., New York City.

SECRETARY.

PROF. F. R. HUTTON.....No. 12 West 31st St., New York City.

NOTE.

THE considerable bulk of the annual volume of Transactions has induced the Publication Committee to direct the insertion of a summary of the Society membership in place of the complete list of members which was published in earlier volumes. The summary attaching to this issue is that which appears in the catalogue of the Society, issued with corrections up to July, 1899. Reference for the complete list should be made to the XXth catalogue issued on that date. The summary is as follows:

FOREIGN COUNTRIES.

	Membership.		Membership.		Membership.
Belgium	2	Great Britain (England)	28	Mexico	3
Canada	7	Great Britain (Scotland)	2	Russia	2
Central America	1	Holland	2	South Africa	2
China	2	India	1	South America	5
Cuba	1	Japan	5	Sweden	5
France	8			Switzerland	1
Germany	6				

Total foreign membership 83

UNITED STATES.

	Membership.		Membership.		Membership.
Alabama	1	Kansas	1	New York	530
Alaska	1	Kentucky	2	North Carolina	3
Arizona	1	Louisiana	4	Ohio	122
Arkansas	1	Maine	11	Oregon	1
California	26	Maryland	24	Pennsylvania	289
Colorado	17	Massachusetts	182	Puerto Rico	1
Connecticut	94	Michigan	53	Rhode Island	40
Delaware	9	Minnesota	8	South Carolina	6
District of Columbia	20	Mississippi	1	Tennessee	2
Florida	1	Missouri	25	Texas	3
Georgia	3	Montana	8	Vermont	8
Illinois	128	Nebraska	3	Virginia	26
Indiana	26	Nevada	1	Washington	2
Iowa	4	New Hampshire	11	West Virginia	4
		New Jersey	101	Wisconsin	41

Total membership in the United States 1,845

GEOGRAPHICAL SUMMARY.

Total foreign membership	83
Total membership in United States	1,845
Present address unknown	1
Total membership	1,929

SUMMARY OF MEMBERSHIP BY GRADES.

Honorary members	15
Members	1,416
Associates	121
Junior members	377
Total membership	1,929

Seventy-five members, by payment of the life membership fee of $200, have made themselves life members, but are included in the above list in their respective grades.

PAST OFFICERS.

(EXECUTIVE.)

PRESIDENTS.

R. H. THURSTON (April 7th, 1880—Nov. 3d, 1882), E. D. LEAVITT, JR. (Nov. 3d, 1882—Nov. 3d, 1883), JOHN E. SWEET (Nov. 3d, 1883—Nov. 7th, 1884), J. F. HOLLOWAY * (Nov. 7th, 1884—Nov. 13th, 1885), COLEMAN SELLERS (Nov. 13th, 1885—Dec. 2d, 1886), GEO. H. BABCOCK † (Dec. 2d, 1886—Dec. 1st, 1887), HORACE SEE (Dec. 1st, 1887—Oct. 18th, 1888), HENRY R. TOWNE (Oct. 18th, 1888—Nov. 22d, 1889), OBERLIN SMITH (Nov. 22d, 1889—Nov. 14th, 1890), ROBT. W. HUNT (Nov. 14th, 1890—Nov. 20th, 1891), CHAS. H. LORING (Nov. 20th, 1891—Nov. 29th, 1892), ECKLEY B. COXE‡ (Nov. 29th, 1892—Dec. 4th, 1894), E. F. C. DAVIS§ (Dec. 4th, 1894—Aug. 6th, 1895), CHAS. E. BILLINGS‖ (Aug. 6th, 1895—Dec. 3d, 1895), JOHN FRITZ (Dec. 3d, 1895—Dec. 5th, 1896), WORCESTER R. WARNER (Dec. 5th, 1896—Dec. 3d, 1897), CHARLES WALLACE HUNT (Dec. 3d, 1897—Dec. 2d, 1898).

TREASURERS AND SECRETARIES.

Treasurers.—LYCURGUS B. MOORE (April 7th, 1880—Dec. 2d, 1881), CHAS. W. COPELAND ¶ (Dec. 2d, 1881—Nov. 7th, 1884).

Secretaries.—LYCURGUS B. MOORE (*Acting*, April 7th, 1880—Nov. 4th, 1880), THOS. WHITESIDE RAE ** (Nov. 4th, 1880—March 1st, 1883).

MEMBERS OF PREVIOUS COUNCILS.

VICE-PRESIDENTS.

HENRY R. WORTHINGTON,†† COLEMAN SELLERS, ECKLEY B. COXE,‡ Q. A. GILLMORE, WM. H. SHOCK, ALEX. L. HOLLEY,‡‡ F. A. PRATT, W. P. TROWBRIDGE,§§ E. D. LEAVITT, JR., CHAS. E. EMERY, JOHN FRITZ, HENRY MORTON, WM. METCALF, S. B. WHITING, A. B. COUCH, W. R. ECKHART, J. V. MERRICK, CHARLES W. COPELAND,¶ OLIN LANDRETH, HENRY R. TOWNE, C. H. LORING, HORACE SEE, ALLAN STIRLING, JOS. MORGAN, JR., C. T. PORTER, HORACE S. SMITH, W. S. G. BAKER, H. G. MORRIS, C. J. H. WOODBURY, THOS. J. BORDEN, WM. KENT, CHAS. B. RICHARDS, JOEL SHARP,‖ GEO. W. WEEKS, DE VOLSON WOOD, S. W. BALDWIN, JOHN F. PANKHURST, ALEXANDER GORDON, GEO. I. ALDEN, E. F. C. DAVIS, IRVING M. SCOTT, C. W. HUNT, THOS. R. PICKERING, EDWIN REYNOLDS, C. E. BILLINGS, PERCIVAL ROBERTS, JR., H. J. SMALL, F. H. BALL, JESSE M. SMITH, M. L. HOLMAN, GEO. W. MELVILLE, CHAS. H. MANNING, FRANCIS W. DEAN, E. S. CRAMP, S. T. WELLMAN, and W. F. DURFEE.

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W. P. TROWBRIDGE,§§ T. N. ELY, J. C. HOADLEY,¶¶ WASHINGTON JONES, WM. B. COGSWELL, F. A. PRATT, CHAS. B. RICHARDS, S. B. WHITING, J. F. HOLLOWAY, GEO. W. FISHER, ALLAN STIRLING, GEO. H. BABCOCK, S. W. ROBINSON, JNO. E. SWEET, R. W. HUNT, CHAS. T. PORTER, C. J. H. WOODBURY, W. F. DURFEE, OBERLIN SMITH, C. C. WORTHINGTON, WM. LEE CHURCH, WM. HEWITT, C. H. MORGAN, H. A. HILL, WM. KENT, S. T. WELLMAN, F. G. COGGIN, J. T. HAWKINS, T. R. MORGAN, SR., S. W. BALDWIN, FRED'K GRINNELL, MORRIS SELLERS, FRANK H. BALL, GEO. M. BOND, WM. FORSYTH, JAS. E. DENTON, CARLETON W. NASON, H. H. WESTINGHOUSE, ANDREW FLETCHER, WORCESTER R. WARNER, COLEMAN SELLERS, JR., JAS. M. DODGE, ROBT. FORSYTH, JESSE M. SMITH, CHAS. H. MANNING, C. W. PUSEY, JOHN THOMSON, JOHN B. HERRESHOFF, L. B. MILLER, W. S. RUSSEL, JOHN C. KAFER, CHAS. A. BAUER, ARTHUR C. WALWORTH, NORMAN C. STILES, E. D. MEIER, and GEO. W. DICKIE.

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* Died, Sept. 1, 1896.
§ Died, Aug. 6, 1895.
** Died, May 27, 1893.
§§ Died, Aug. 12, 1892.
† Died, Dec. 16, 1893.
‖ Unexpired term of Mr. Davis.
†† Died, Dec. 17, 1880.
‖‖ Died, July 28, 1898.
‡ Died, May 13, 1895.
¶ Died, Feb. 7, 1895.
‡‡ Died, Jan. 29, 1882.
¶¶ Died, Oct. 21, 1886.

HONORARY COUNCILLORS.

Past Presidents of the Society.

R. H. THURSTON	1880—1882	Ithaca, N. Y.
E. D. LEAVITT	1882—1883	Cambridgeport, Mass.
JOHN E. SWEET	1883—1884	Syracuse, N. Y.
COLEMAN SELLERS	1885—1886	Philadelphia, Pa.
HORACE SEE	1887—1888	New York City.
HENRY R. TOWNE	1888—1889	Stamford, Conn.
OBERLIN SMITH	1889—1890	Bridgeton, N. J.
ROBERT W. HUNT	1890—1891	Chicago, Ill.
CHARLES H. LORING	1891—1892	Brooklyn, N. Y.
CHARLES E. BILLINGS*	1895	Hartford, Conn.
JOHN FRITZ	1895—1896	Bethlehem, Pa.
WORCESTER R. WARNER	1896—1897	Cleveland, Ohio.
CHARLES WALLACE HUNT	1897—1898	New York City.

[NOTE.—The former Presidents of the Society are members of the Council for life or during their retention of active membership in the Society.]

* Unexpired term of E. F. C. Davis.

RULES OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

ART. 1. The objects of the AMERICAN SOCIETY OF MECHANICAL ENGINEERS are to promote the Arts and Sciences connected with Engineering and Mechanical Construction, by means of meetings for social intercourse and the reading and discussion of professional papers, and to circulate, by means of publication among its members, the information thus obtained.

ART. 2. All persons connected with engineering may be eligible for admission into the Society.

ART. 3. The Society shall consist of Honorary Members, Members, Associates, and Juniors.

ART. 4. Honorary Members, not exceeding twenty-five in number, may be elected. They must be persons of acknowledged professional eminence.

ART. 5. To be eligible as a Member, the candidate must be not less than thirty years of age, and must have been so connected with engineering as to be competent as a designer or as a constructor, or to take responsible charge of work in his department, or he must have served as a teacher of engineering for more than five years.

NOTE.—The Rules of the Society, adopted in 1880, were in force until 1884, when they received general revision by a careful committee, whose report, distributed by letter ballot, was adopted November 5, 1884. In December, 1894, a similar extensive revision was made under direction of the Council, and the present rules are those of 1894. They include the amendments made in 1889, 1891, 1893, and 1898, which were the only changes since the revision of 1884.

ART. 6. To be eligible as an Associate the candidate must be not less than twenty-six years of age, and must have the other qualifications of a member; or he shall have been so connected with engineering as to be competent to take charge of work, and to coöperate with engineers.

ART. 7. To be eligible as a Junior, the candidate must have had such engineering experience as will enable him to fill a responsible position, or he must be a graduate of an engineering school.

ART. 8. All Honorary Members, Members, and Associates shall be equally entitled to the privileges of membership. Juniors shall not be entitled to vote, nor to be officers of the Society.

ART. 9. Nominees for Honorary Membership must be proposed by at least five Members who are not officers of the Society. References shall not be required of a nominee for Honorary Membership, but the grounds upon which the application is made must be fully set forth in writing and signed by the proposers.

ART. 10. A candidate for admission to the Society, as a Member or as an Associate, must make an application on a form to be prepared by the Council, which shall contain a written statement giving a complete account of his engineering experience and an agreement that he will, if elected, conform to the laws, rules, and requirements of the Society. He must refer to at least five Members or Associates to whom he is personally known. A candidate for admission to the Society as a Junior must make an application on the same form, and refer to not less than three Members or Associates to whom he is personally known.

Applications for membership from engineers who are not resident in the United States and Canada, and who may be so situated as not to be personally known to five Members of the Society, as required in the foregoing paragraph, may be recommended for ballot by five Members of the Council, after sufficient evidence has been secured which shall show that in their opinion the applicant is worthy of admission to the grade which he seeks.

ART. 11. The referees for each candidate for admission to the Society shall be requested to make a confidential communication on a form to be prepared by the Council, setting forth in detail

such information, personally known by the referee, as shall enable the Council to arrive at a proper estimate of the eligibility of the candidate for admission to the Society. Such confidential communications shall be destroyed by the Secretary as soon as the vote has been officially declared.

Art. 12. All applications for membership must be presented to the Council, and this body shall consider each application, assigning to each, with the applicant's consent, the grade in the Society to which, in its opinion, his qualifications entitle him. The names of those candidates recommended for election by the Society shall be immediately printed on a ballot, and the ballot mailed at once by the Secretary to each voting member of the Society. Persons desiring to change their grade of membership from junior to associate or from associate to member shall make an application in the same manner and on the same form as that required for a new applicant.

Art. 13. A member entitled to vote may leave the name of any candidate on the ballot untouched to vote in favor of the admission of the candidate to the Society, or he may erase the name to vote against it. He shall enclose the ballot so approved by him in a sealed blank envelope, and enclose this envelope in a second envelope, on which he shall write his name, and mail the same to the Secretary of the Society. A ballot without such endorsement shall be rejected as defective. The rejection of a candidate by seven voters shall defeat his election.

Art. 14. The aforesaid envelopes containing the ballots shall be opened by the Council, at any meeting thereof, and the names of those elected shall be announced in the next meeting of the Society. The names of applicants not elected shall not be announced, nor recorded in the proceedings.

Art. 15. Endorsers of any applicant not elected may, within three months after such failure to be elected, lay before the Council written evidence that an error was then made. The Council may then, by a three-fourths vote, order another similar ballot by the Society, in which case thirteen negative votes shall be required to defeat the candidate.

Art. 16. Honorary members shall be elected by the unanimous vote of the Council, through a letter ballot, not less than sixty days subsequent to the proposal, a notice of which proposed election shall have been mailed at once by the Secretary to each member of the Council.

ART. 17. Each person elected, excepting honorary members, must subscribe to the Rules of the Society, and pay the initiation fee before he can receive a certificate entitling him to the rights and privileges of the Society, and to wear the emblem appropriate to his grade. If this payment is not made within six months of the election, the same shall be void, unless the time is extended by the Council. The emblems of each grade of membership shall be worn by those only who belong to that grade.

ART. 18. The initiation fee of a member or an associate shall be twenty-five dollars, and the annual dues shall be fifteen dollars, payable in advance. The initiation fee of a junior shall be fifteen dollars, and his annual dues ten dollars, payable in advance. A junior being promoted to any other grade of membership shall pay an additional initiation fee of ten dollars. Any member or associate may become a Life Member in the same grade, by the payment of two hundred dollars at one time, and shall not be liable thereafter to annual dues.

The Council shall have the power, for special reasons, by unanimous vote, through a letter ballot, to admit to life membership, without the payment of the sum above named, such person as for a long term of years has been a member or an associate, when such a procedure would in its judgment be for the best interests of the Society; *provided*, that notice of such action shall have been given at a previous meeting of the Council.

ART. 19. Any member of the Society in arrears may, at the discretion of the Council, be deprived of the publications of the Society, or, when in arrears for one year, he may be stricken from the list of members. Such person may be restored to the privileges of membership by the Council on payment of all arrears.

ART. 20. The affairs of the Society shall be managed by a Council, consisting of a President, six Vice-Presidents, nine Managers, and a Treasurer, who shall also be the Trustees of the Society.

All past (ex) Presidents of the Society, while they retain their membership therein, shall be known as Honorary Councillors, and shall be entitled to receive notices of all meetings of the Council and may take part in any of its deliberations; they shall be entitled to vote upon all questions except such as affect the legal rights or obligations of the Society or its members.

ART. 21. The members of the Council shall be elected from among the members and associates of the Society at the annual meetings, and shall hold office as follows:

The President and the Treasurer for one year; and no person shall be eligible for immediate re-election as President who shall have held that office for two consecutive years; the Vice-Presidents for two years, and the Managers for three years; and no Vice-President or Manager shall be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

ART. 22. A Secretary, who shall be a member of the Society, shall be appointed for one year by a majority of the members of the Council at its first meeting after the annual election, or as soon thereafter as the votes of a majority of the members of the Council can be secured for a candidate. The Secretary may be removed by a vote of twelve members of the Council, at any time after one month's notice has been given him by a majority of its members to show cause why he should not be removed, and he has been heard to that effect. The Secretary may take part in any of the deliberations of the Council, but shall not have a vote therein. His salary shall be fixed for the time he is appointed by a majority vote of the Council.

ART. 23. At each annual meeting, a President, three Vice-Presidents, three Managers, and a Treasurer shall be elected, and the term of office of each shall continue until the end of the meeting at which their successors are elected.

ART. 24. The duties of all officers shall be such as usually pertain to their offices or may be delegated to them by the Council or by the Society. The Council may, in its discretion, require bonds to be given by the Treasurer.

ART. 25. The Council may, by vote of a majority of all its members, declare the place of any officer vacant, on his failure for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office. All such vacancies and those occurring by death or resignation shall be filled by the appointment of the Council, and any person so appointed shall hold office for the remainder of the term for which his predecessor was elected or appointed; *provided*, that the said appointment shall not render him ineligible at the next annual meeting.

ART. 26. Five members of the Council shall constitute a quorum. Members of the Council absent from a meeting may vote by letter upon subjects stated in the call for the meeting, said vote to be deposited with the Secretary.

ART. 27. The President on assuming office shall appoint a Finance Committee and a Publication Committee and a Library Committee of five members each. The appointment of two members of each Committee shall expire at the end of each year. The Secretary shall, *ex officio*, be a member of all three committees.

ART. 28. The Finance Committee shall have power to order all ordinary or current expenditures, and shall audit all bills therefor. No bill shall be paid except upon their audit. When special appropriations are ordered by the Society, they shall not take effect until they have been referred to the Council and Finance Committee in conference.

ART. 29. It shall be the duty of the Publication Committee to receive all papers contributed, and to decide upon which papers or parts of the same shall be presented at the professional meetings of the Society. They shall see that all editorial revisions of the proceedings, papers, discussions, and reports are made; and to decide what parts of the same shall be published in the proceedings of the Society. The Council may, at its discretion, revise any action of the Publication Committee.

ART. 30. It shall be the duty of the Library Committee to take charge of the collection of all material for the Library of the Society, and to supervise all regulations for its use.

ART. 31. At the regular meeting preceding the annual meeting a Nominating Committee of five members, not officers of the Society, shall be appointed, and this committee shall, at least thirty days before the annual meeting, send to the Secretary the names of nominees for the offices falling vacant under the rules. In addition to such regularly appointed committee, any other five members or associates, not in arrears, may constitute an independent Nominating Committee, and may present to the Secretary, at least thirty days before the annual meeting, all the names of such candidates as they may select. All the names of such independent nominees shall be placed upon the ballot list, with nothing to distinguish them from the nominees of the regular committee, and the Secretary shall at once mail the said list of names to each member and associate in the form of a letter ballot, it being understood that the assent of the nominees shall have been secured in all cases.

ART. 32. In the election of Vice-Presidents, each member and associate may cast as many votes as there are Vice-Presidents

to be elected. He may give all these votes to one candidate, or distribute them among more, as he chooses. Managers shall be voted for in the same way.

ART. 33. Any member or associate entitled to vote may vote by retaining or changing the names on said list, leaving names not exceeding in number the officers to be elected, and returning the list to the Secretary—such ballot enclosed in two envelopes, the inner one to be blank and the outer one to be endorsed by the voter. No member or associate in arrears since the last annual meeting shall be allowed to vote until said arrears shall have been paid.

ART. 34. The said blank envelopes shall be opened by tellers at the annual meeting, and the person who shall have received the greatest number of votes for the several offices shall be declared elected.

MEETINGS.

ART. 35. The annual meeting of the Society shall be held on the first Tuesday in December of each year, in the City of New York, unless otherwise ordered, at which a report of proceedings and an abstract of the accounts shall be furnished by the Council. The Council may change the place of the annual meeting, and shall, in that case, give timely notice to members and associates.

ART. 36. Other regular meetings of the Society shall be held in each year at such time and place as the Council may appoint. At least thirty days' notice of all meetings shall be mailed by the Secretary to members, honorary members, associates, and juniors.

ART. 37. Special meetings may be called whenever the Council may see fit; and the Secretary shall call a special meeting at the written request of twenty or more members. The notices for special meetings shall state the business to be transacted, and no other shall be entertained.

ART. 38. Any member, honorary member, or associate, may introduce a stranger to any meeting; but the latter shall not take part in the proceedings without the consent of the meeting.

ART 39. Every question which shall come before the Society shall be decided, unless otherwise provided by these rules, by the votes of a majority of the members and associates present, provided there is a quorum.

ART. 40. At any regular meeting of the Society thirteen or more members and associates shall constitute a quorum.

ART. 41. Unless otherwise ordered, papers shall be read in the order in which their text is received by the Secretary. Before any paper appears in the *Transactions* of the Society, a copy of the paper shall be sent to the author, and, so far as possible, a copy of the reported discussion shall be sent to every member who took part in the same, with requests that attention shall be called to any errors therein.

ART. 42. The Society shall claim no exclusive copyright in papers read at its meetings, nor in reports of discussions, except in the matter of official publication with the Society's imprint, as its *Transactions*. The Secretary shall have sole possession of papers between the time of their acceptance by the Publication Committee and their reading, together with the drawings illustrating the same; and at the time of such reading, or as soon thereafter as practicable, he shall cause to be printed, with the authors' consent, copies of such papers, "subject to revision," with such illustrations as are needed for the *Transactions*, for distribution to the members and for the use of technical newspapers, American and foreign, which may desire to reprint them in whole or in part. The policy of the Society in this matter shall be to give papers read before it the widest circulation possible, with the view of making the work of the Society known, encouraging mechanical progress, and extending the professional reputation of its members.

ART. 43. The author of each paper read before the Society shall be entitled to twelve copies, if printed, for his own use, and all members shall have the right to order any number of reprints of papers at a cost to cover paper and printing; *provided*, that said copies are not intended for sale.

ART. 44. The Society is not, as a body, responsible for the statements of fact or opinion advanced in papers or discussions, at its meetings; and it is understood that papers and discussions should not include matters relating to politics or purely to trade.

ART. 45. These rules may be amended, at any annual meeting, by a two-thirds vote of the members present; *provided*, that written notice of the proposed amendment shall have been given at a previous meeting.

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PAPERS

OF THE

NEW YORK MEETING

(XXXVIIIth)

NOVEMBER 29th TO DECEMBER 2d, 1898.

DCCXCI.

PROCEEDINGS

OF THE

NEW YORK MEETING

(XXXVIIIth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

November 29th to December 2d, 1898.

THE nineteenth annual meeting of the Society, which was also its thirty-eighth convention, was held in New York City, during the interval, November 29 to December 2, 1898. The formal gatherings were held in the Society's cozy auditorium, but a large overflow gathering was always to be found in the halls and parlors, of those who prized the opportunity for renewing old acquaintances and making new ones.

The first session opened on the evening of Tuesday, November 29th, by the gathering of the members in the parlors and library for a species of informal reception. At about nine o'clock the audience was convened in the auditorium to listen to the address of the President, Mr. Charles Wallace Hunt of New York, which was heard with profound interest and attention. His subject was, "The Engineer: His Work; His Ethics; His Pleasures."

Messrs. Bonner and Barr were appointed tellers, to count the letter ballot received from the membership for officers of the Society for the ensuing year, to render a report at the business session on Wednesday morning, and after announcements concerning the conduct of the meeting, the members adjourned for a light luncheon in the banquet room below.

Second Day. Wednesday, November 30th.

The regular business session of the annual meeting of the Society convened at ten o'clock, in the auditorium of the Society. The large number already registered indicated a considerable numerical success for the meeting, although the severe weather and storms in New England materially affected the attendance of members from that region. The Society was saddened to learn of the death, by the loss of the *City of Portland* at sea, of Mr. Wm. L. Chase, one of its valued members.

The register of members and guests contained the following names of members present at the meeting:

Albree, C. B.
Allan, Percy.
Aller, A.
Allison, Robt.
Almirall, J. A.
Almond, Thos. R.
Almy, Darwin.
Archer, E. R.
Ashworth, D.
Aslakson, B.
Bailey, W. H.
Ball, F. H.
Bang, H. A.
Baldwin, S. W.
Barnard, G. A.
Barnes, A. T.
Barth, C. G.
Barr, J. H.
Barr, H. P.
Barrus, Geo. H.
Basford, G. M.
Bates, E. P.
Bates, F. C.
Bauer, C. A.
Baylis, R. N.
Beach, G.
Bellows, L. E.
Benns, C. P.
Billings, C. E.
Bird, W. W.
Birkinbine, J.
Blessing, J. H.
Bole, W. A.
Bolton, R. P.
Bond, G. M.
Bonner, W. T.
Boyer, E. S.
Bradley, W. H.
Brashear, J. A.
Bristol, Wm. H.
Brown, R. S.
Brown, Wm. Clinton.
Bulkley, H. W.
Burdett, W. F.
Burnham, H. A.
Burns, A. L.
Caldwell, A. J.
Camp, Geo. E.
Carpenter, A. H.
Cary, A. A.
Cassier, Louis.
Chambers, F. R., Jr.
Cheney, W. L.
Christensen, A. C.
Christie, W. W.
Church, E. D., Jr.
Clarke, S. J.
Cobb, G. H.
Cogswell, W. B.
Colvin, F. H.
Colwell, A. W.
Conant, Wm. S.
Connell, J. A.
Connolly, J. A.
Conrad, H. V.
Cook, Harte.
Corbett, C. H.
Coster, E. L.
Cox, J. D., Jr.
Cremer, J. M.
Cruikshank, L. B.
Cullingworth, G. R.
Dale, O. G.
Dallett, W. P.
Darling, E. A.
Davis, L. K.
Deane, C. P.
Decrow, D. A.
Deming, W. H.
Detrick, J. S.
Dinkel, Geo.
Donnelly, J. R.
Ducommun, Ed.
Durfee, W. F.
Ekstrand, Chas.
Evans, Q. N.
Faber du Faur, A.
Fellows, E. R.
Ferguson, G. R.
Finley, A. D.
Fladd, F. C.
Flint, B. P.
Fowler, G. L.
Francis, H. C.
Freeman, J. R.
Frevert, H. F.
Frith, A. J.
Fritz, John.
Frothingham, F. F
Gantt, H. L.
Gobeille, J. L.
Goetz, V. J.
Goetze, F. A.
Goodale, A. M.
Gould, W. V.
Gowing, E. H.
Granger, A. S.
Green, Saml. M.

Greenleaf, G. E.
Greensmith, Jas. E.
Greer, R. C.
Grohmann, C. L.
Gulowsen, G. A.
Hale, R. S.
Hall, F. A.
Hall, John H.
Hall, R. E.
Halsey, F. A.
Hamilton, J. V.
Henderson, Alex.
Henning, G. C.
Henderson, J. C.
Henry, W. T.
Hewlett, E. M.
Higgins, C. P.
Hill, Geo.
Hill, G. H.
Hillman, Gustave.
Hill, W. E.
Hill, Wm.
Hoffecker, W. L.
Holly, E. P.
Holmes, W. G.
Hoppes, J. J.
Hopton, W. E.
Horton, J. T.
Hough, D. L.
Hunt, A. E.
Hunt, Chas. Wallace.
Hunt, R. W.
Hunt, W. F.
Huson, W. S.
Huston, C. L.
Hutton, F. R., Sec'y.
Idell, F. E.
Inglis, Jas.
Jacobi, A. W.
Jacobus, D. S.
Jenkins, M. C.
Jenks, L. H.
Jenks, Wm. H.
Jennings, E. L.
Johnson, A. E.
Jordan, S. S.
Kafer, J. C.
Katte, E. B.
Kent, Wm.
King, C. C.
Kingsbury, Albert.
Kirchhoff, Chas.
Laforge, F. H.
Langlotz, Chas.
Langlotz, Robt.
Lawton, M. A.
LeVan, W. B.
Lenssen, Arthur, Jr.
Lewis, D. J., Jr.
Lewis, J. F.
Lieb, J. W., Jr.
Lipps, H., Jr.
Longnecker, C. K.
Loring, Chas. H.
Loveland, J. W.
Low, F. R.
Luther, F. S.
McArthur, Robt.
McBride, Jas.
McEwen, J. H.
McKean, R. A.
McMannis, Wm.
Mackintosh, Fred'k.
Martinez, M. J.
Matlack, J. R., Jr.
Mattes, W. F.
Massey, G. B.
Mead, F. S.
Meatz, J. T.
Meier, E. D.
Melville, Geo. W.
Melvin, D. N.
Meyer, H. C., Jr.
Miller, F. J.
Moeller, F.
Monaghan, W. F.
Moore, D. G.
Morison, G. S.
Morris, H. G.
Morse, C. M.
Moulthrop, L.
Muller, M. A.
Muller, T. H.
Mumford, E. A.
Nason, C. W.
Neff, E. H.
Newcomb, Chas. L.
Newhall, J. B.
Nordland, Lawrence.
Norris, J. H.
Norris, H. M.
Norris, R. Van A.
Odell, W. H.
Olin, F. W.
Orr, A. M., Jr.
Oviatt, D. B.
Parsons, F. W.
Parsons, H. de B.
Patterson, A. W.
Paul, J. W.
Peek, G. M.
Perkins, T. C.
Pessano, A. C.
Phelps, F. A., Jr.
Phillips, Franklin.
Philip, C. von.
Phipps, C. W.
Platt, John.
Poole, Herman.
Poore, J. B.
Powell, Marcus.
Pratt, C. R.
Prescott, J. A.
Rankin, T. L.
Reed, S. G.
Reist, H. G.
Rice, A. L.
Richards, C. R.
Richards, Frank.
Richards, F. H.
Richmond, Geo.
Ridsdale, T. W.
Robinson, Edw.
Robinson, H. B.
Robinson, J. M.
Robinson, W. S.
Rockwood, G. I.
Rogers, W. S.
Rohrer, A. L.
Roney, Wm. R.
Rood, V. H.
Ross, E. L.
Rowland, A. E.
Rowland, C. B.
Rowland, T. F.
Russell, C. M.
Sabine, A. H.
Sanguinetti, Percy A.
Sattler, Wm. R.
Scheffler, F. A.

Scholl, J.
Schnuck, E. F.
Schoenborn, W. E.
Schumann, F.
Schutte, L.
Scott, Seaton M.
See, Horace.
Sergeant, C. H.
Serrel, J. A.
Sewall, M. W.
Shipley, Thos.
Sinclair, Angus.
Slater, F. R.
Stetson, G. R.
Stratton, W. H.
Smith, A. W.
Smith, E.
Smith, Howard Wells.
Smith, Jesse M.
Smith, J. Waldo.
Smith, Oberlin.
Smith, S. Howard.
Snell, Henry I.
Souther, Henry.
Sparrow, E. P.
Spaulding, H. C.
Spies, Albert.
Spilsbury, E. G.
Stangland, B. F.
Stanley, A. W.
Stearns, Albert.
Stiles, N. C.
Stillman, F. H.
Suplee, H. H.
Svenson, John.
Swasey, Ambrose.
Sweet, J. E.
Thomas, C. W.
Thompson, Ed. P.
Thomson, John.
Thorpe, R. H.
Tibbals, G. A.
Tolman, Jas. P.
Tompkins, Stonewall.
Tompkins, S. D.
Torrance, Kenneth.
Torrey, H. G.
Towle, F. M.
Trask, G. F. D.
Trautwein, A. P.
Treat, Chas. H.
Tremaine, E. G.
Tucker, E. D.
Tucker, W. B.
Turner, John.
Tyler, C. C.
Van Derhoef, G. N.
Varney, W. W.
Veeder, C. H.
Verastegui, Albert.
Wagner, E. F.
Wallace, F. A.
Walworth, A. C.
Ward, W. E.
Warner, Worcester R.
Warren, B. H.
Washington, W. D. H.
Webb, J. B.
Webber, S. S.
Weber, F. C.
Weber, Geo. A.
Webster, W. R.
Wellman, S. T.
Wheeler, F. M.
Wheelock, J.
Whinery, S.
Whitney, W. M.
Whittier, Chas.
Wiggin, Wm. H.
Wiley, Wm. H.
Williamson, Wm. C.
Williams, Franklin.
Williston, Arthur.
Winship, J. G.
Wood, A. C.
Wood, A. J.
Woodbury, C. J. H.
Woolson, O. C.
Wright, J. K.
Wurts, Edw. B.
Wyman, H. W.
Young, W. S.

There was also a considerable number of invited guests and ladies.

The plan was followed of numbering the signatures on the official register, and providing that a lapel-badge worn at the convention should bear the number corresponding to the signature upon the register. Transcripts from the register were printed at short intervals and distributed, so that every facility was furnished for the members to become acquainted with each other. It was remarked that the social side of the meeting has appeared to grow stronger and stronger as an element of its attraction, the longer this plan is in operation.

The first business of the general session were the reports required under the Rules, from the Council and its Standing Committees. These had been printed and were read in abstract by the Secretary. Their full text is as follows:

ANNUAL REPORT OF THE COUNCIL.

The Council would present to the Society, convened for its XIXth Annual Meeting, the report of business which has been considered by it and of the action which has been taken during the year.

Pursuant to the resignation of Mr. David R. Frazer, elected Vice-President of the Society by the ballot counted at the Annual Meeting of 1897, the Council has appointed, under Article 25 of the Rules, Mr. Charles M. Jarvis to act as Vice-President and member of the Council for the full term of two years, for which his predecessor had been elected.

There was referred to the Council by the Annual Meeting of 1897 the question of appointing a committee to consider the subject of Standard Pipe Unions, and to confer with similar committees of the American Railway Master Mechanics and Master Car Builders' Asssociation. Under this resolution the Council appointed a committee of five, being the same committee who was appointed in 1885 to consider the question of Standardization of Pipe Threads. Resignations from that committee, by reason of business changes or absences from the country, resulted in the appointment finally of a different committee, consisting of Messrs. B. H. Warren, Chairman; Wm. J. Baldwin, George M. Bond, E. M. Herr, and Stanley G. Flagg. The work of this committee has been so interfered with by the assignments of its members to military duty or naval appointments during the war that the committee has no report to make at the present meeting.

The following letter has been received from the Institution of Civil Engineers of Great Britain:

"To the President and Council of the American Society of Mechanical Engineers.

"Gentlemen: In view of the intended International Exposition in Paris in the year 1900, it has appeared to the Council of this Institution not unlikely that some of the members of your Society may visit Europe in a more or less organized party in that year.

"We are desired by the Council of this Institution to say, that should such a step be taken, and should your members be able to visit England, the Institution of Civil Engineers would wish to welcome its professional brethren of the United States with a warm greeting, to receive them at the house of the Institution, and in such other ways as may be found agreeable, to take advantage of such a favorable opportunity of testifying their regard for the members of your Society.

"We have the honor to be,

"Gentlemen, yours very truly,

"J. Wolfe Barry, *President.*

"J. H. T. Tudsbery, *Secretary.*"

The Secretary of this Society was directed to write to the Council of the Institution, acknowledging the receipt of the invitation, and stating that a fitting response would be made in due time, after the matter can be thoroughly canvassed among the membership.

The Council has directed that the Society should be represented in the conference proposed under the auspices of the International Association for Testing Materials. The Society has been made a member of this organization, by the payment of its yearly fee.

A Committee has been appointed to consider and report upon the methods pursued in the large professional societies of England and the Continent in the matter of electing members of these societies in their several grades. It has been the opinion of the Council that the growth of the Society from the comparatively small beginnings and a limited membership to the present international membership makes it desirable that the veto power upon the candidacy of engineers to membership in the Society should not vest in the possession of a percentage of the Society, which is continually growing smaller and smaller. The whole question, therefore, of the wise procedure, and of the amendment to the present rules which may be desirable to carry out this wise course, is under consideration by the Council, but no report is ready for this meeting.

Pursuant to the papers presented by Messrs. Bryan Donkin, of London, England, and Mr. George H. Barrus, of Boston, and the resolution of the Niagara Falls Convention that a committee of five should be appointed to consider and report upon a "Standardization or Unification of the Methods of Testing Steam-engines," the Council has approved the appointment of a committee consisting of Messrs. Francis H. Boyer, George H. Barrus, Bryan Donkin, D. S. Jacobus, and George H. Richmond.

The Committee of the Society, consisting of Messrs. H. de B. Parsons and T. F. Rowland, Jr., which was appointed to serve with a similar delegation from other organizations to conduct a series of tests upon the fire-resisting properties of fire proofing materials, has handed in its resignation. The exhaustion of the funds available for the conduct of these tests, and changes in the conditions attendant upon them, has made it impossible for the Committee to prosecute further researches.

The Council has had to consider an application from a mem-

ber elected to the Member's Grade, who, after such election but before qualifying, had expressed a desire to change his grade of membership to a lower grade. While the reasons advanced for this desire were cogent, mainly on their financial side, it was the opinion of the Council that the best interests of the Society would be subserved by having in each grade of membership only such engineers as were qualified in the opinion of the Council to attach to that grade. It is the expectation of the Council, and the implied provision of the rules, that when a person by reason of age or experience has become capable of passing from the lower grade to the higher that he should take steps to secure this result.

The Council would report for record in its *Transactions* the following lists of deaths since the publication of its annual report in December, 1897: G. J. Maillefert, December 9, 1897; B. K. Field, January 13; W. E. Stearns, May 6; Charles E. Emery, June 1; Edwin H. Bennett, June 27; Joel Sharp, July 28; J. J. Schoenleber, August 4; W. H. Inslee, July 24; Henri Adolph E. Schneider, May 17; John C. O'Connell, July 12; Joseph C. Platt, July 7; F. F. Hemenway, October; H. B. Miller, October; Joshua Rose, November 15, 1898.

The Council would report the following summary of the membership of the Society at the Annual Meeting of 1898. The list contains those who have been elected preceding this meeting, and excludes those who by prolonged failure to pay the dues of the Society have allowed their membership to lapse under the provisions of the Rules:

Honorary members	15
Members	1,381
Associates	118
Junior members	347
Total	1,861

The Council would also present the report of its tellers appointed to count the ballots cast for membership in the Society at the Annual Meeting.

REPORT OF TELLERS OF ELECTION.

The undersigned were appointed a committee of the Council to act as tellers, under Article 14 of the Rules, to scrutinize and count the ballots cast for and against the candidates proposed for membership in their several grades in the American Society

of Mechanical Engineers, and seeking election before the XXXVIIIth Meeting, New York, 1898.

They have met upon the designated day, in the office of the Society, and have proceeded to the discharge of their duty. They would certify, for formal insertion in the records of the Society, to the election of the following persons whose names appear on the appended list, in their several grades.

There were 471 votes cast on the pink ballot, of which 10 were thrown out because of informalities. The tellers have considered a ballot as informal which was not indorsed with an autographic signature, or where the indorsement was made by a facsimile or other stamp.

CHAS. H. LORING,
H. H. SUPLEE,
GUS. C. HENNING,
GEO. RICHMOND.
} *Tellers of Election.*

AS MEMBERS.

Bolton, R. P.
Dawson, Philip.
Evans, Geo. H.
Fellows, Edwin R.
Hamilton, James.
Hewlings, Andrew J.
Johnson, Frederic A.
Johnson, Wallace C.
Locke, Wm.
McGuire, Jno. P.
Pierce, Edward L.
Poore, J. B.
Robinson, Jno. C.
Root, Orlando J.
Saunders, Augustus M.
Snow, Walter B.
Stephens, Geo.
Tejada, Juan de D.
Warner, Adana D.
Wilkin, Jno. T.

AS ASSOCIATES.

Bacon, Geo. W.
Cushman, Herbert E.
Smith, Louis L.
Tompkins, Saml. D.

PROMOTION TO FULL MEMBERSHIP.

Bonner, Wm. T.
Van Derhoef, Geo. N.

AS JUNIOR MEMBERS.

Finley, A. D.
Hall, Robt. E.
Holloway, Carl S.
Miller, Robt. S.
Morrison, H. H.
Patton, John H.
Richmond, Harold A.
Schaake, William.
Schreuder, Andrew M.
Villere, St. Denis, J.
Walker, Fredk. W.
White, Harry C.
Wood, Arthur J.

At the close of the report of the Council, the second order of business was the Report of the Finance Committee, which was as follows:

ANNUAL REPORT OF THE FINANCE COMMITTEE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 1897–1898.

For the fiscal year, 1897–8, the Finance Committee of the American Society of Mechanical Engineers would respectfully report to the Council and the Society the following statements of receipts and expenditures which have passed under their direction on behalf of the Society during the year beginning November 15, 1897, and ending November 15, 1898.

Secretary's Balance Sheet for the fiscal year ending November 15, 1898:

Dr.		Cr.	
To receipts for the year	$32,406 87	By Cash to Treasurer	$32,406 87

Itemized statement of receipts and expenditures of the Society for fiscal year 1897–1898:

Receipts.

Dr.	
Initiation Fee	$2,585 00
Current Dues	22,351 54
Past Dues	1,100 78
Advance Dues	186 17
Sales of Publications	1,122 22
Binding	26 75
Engraving	152 49
Life Membership (cash)	784 97
Interest on Investment	1,180 00
Office Expenses	10 38
Mail and Express	1 13
Badges	577 50
Hall Rentals	595 00
Room "	1,635 81
Insurance	10 86
Certificates	2 00
Library	55 00
Traveling	11 70
Rent, Interest and Taxes	02
Lighting	2 50
House Supplies and Furniture	14 15
Janitorial Supplies	90
Total Receipts	$32,406 87
Cash on hand first of year	47 60
Total	$32,454 47
Cash on hand, forward	$698 65

Disbursements.

Cr.	
Reprints and Publications	$8,863 28
Postage and Express	1,787 70
Salaries	7,365 00
Office Expenses	308 98
Engraving	834 29
Contingencies (profit and loss)	15 25
Binding *Transactions*	1,667 05
Meetings	798 47
Work of Committees	464 18
Badges and Certificates	587 66
Traveling	126 35
Insurance and Safety Deposit	40 00
Rent, Interest, and Taxes	4,509 35
Printing, Circulars, Catalogues, Office Forms, etc.	1,599 00
Stationery Supplies	365 34
House Supplies and Furnitnre	656 07
Library (book purchase and binding)	178 85
Janitorial Supplies	131 40
Fuel	198 25
Lighting (gas and electric light)	372 14
Laundry	200 00
Repairs to House Furniture, etc	672 21
Dues (repayment to a member)	15 00
Total Disbursements	$31,755 82
Cash on hand to balance	698 65
Total	$32,454 47

Of the issue of bonds, in 1890, of the Mechanical Engineers' Library Association, which amounted to $32,000, the Council of the Society, as Trustees, held November 15th, 1897, at the time of the last report of the Finance Committee, $23,400, and during the year 1897–98, the Council has acquired $200 additional bonds in payment for a Life Membership, thus making $23,600, so held by the Council at this date, and leaving $8,400 of said bonds still outstanding in the hands of the members.

At the time of this report there remains outstanding uncollected accounts due the society at the end of the year 1897–98 as follows :

171 members (less than 9 per cent. of the entire membership) owe for dues, publications, etc.	$4,368 98
9 men owe for room rent, all recent accounts	55 00
1 society owes for Hall rent, " "	25 00
8 non-members owe for publications purchased	31 60
2 " " " " electros furnished them	5 65
Total uncollected	$4,486 23

Of the 171 members who owe $4,368.98, 98 men owe for dues of the year 1897–98 only, 33 owe dues for two years, one man owes for a badge, and the balance 39 men owe for over three years' dues.

Out of the 171 delinquents, 62 men have had personal letters written them, and circular letters have been sent the rest and replies have been received from 56 saying they would remit by a fixed date, or shortly, or else for valid reasons asking for an extension of time to meet their indebtedness, which has been granted.

This leaves only 115 men out of a membership of 1,844 who have failed either to pay their dues or to write us in respect to meeting the account against them.

The Finance Committee takes pleasure in reporting that the outstanding indebtedness of $3,200.60, which the Society owed at the time of the last report has been entirely wiped out during the year just closed, and further, that all the bills of the year 1897–98 have been paid in full with the exception of $485 due the printer of the *Transactions*, with whom we have a running account, and which sum is more than covered by the cash balance on hand of $698.65 reported above.

ASSETS AND LIABILITIES.

At the end of the year 1897–98.

Exclusive of such property as pamphlet copies of the papers, office furniture, etc., which is also owned by the Society.

Assets.

Bonds of the M. E. L. A. held by the Council as Trustees	$23,600 00
Stock of bound and paper copies of the Society's *Transactions*, Vols. I-XVIII, inclusive, on hand, estimating bound volumes at $6.00 each and paper bound at $5.00 each, the lowest prices at which they are ever sold	24,107 00
Outstanding indebtedness due the Society as itemized above, $4,486.23 estimating 75 per cent. as collectible, which is a safe estimate	3,364 67
Cash on hand and in Bank as reported above	698 65
Total Assets	$51,770 32

Liabilities.

Bill owed printer of *Transactions*, as explained above	485 00
Excess Assets over Liabilities	$51,285 32

The above statement of assets, moreover, does not cover the stereotype plates of the volumes of *Transactions*, nor the electrotypes of the cuts used in illustrating them. It covers only assets which are convertible under favorable conditions into cash. Nor does it include office furniture and fixtures in use.

MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

COPY OF THE ANNUAL REPORT OF THE TRUSTEES OF THE MECHANICAL ENGINEERS' LIBRARY ASSOCIATION, 1897–1898.

The summary of receipts and disbursements of the Trustees from November 16, 1897, to November 15, 1898, is appended.

Secretary's Balance Sheet year 1897–1898:

Dr.		Cr.	
To balance on hand first of year	$2 62	By expenditures as itemized below	$3,977 41
To receipts as itemized below	5,168 88	" cash on hand	1,194 09
	$5,171 50		$5,171 50
To cash on hand	$1,194 09		

Itemized Statement of Receipts and Expenditures of the Association for Fiscal Year 1897–1898 :

Dr.		Cr.	
Receipts, fellowship fund	$184 00	Interest on mortgage	$1,402 50
" sinking fund	426 50	" " bonds	1,600 00
" insurance (refund)	10 18	Salaries	780 00
" office rent	4,500 00	Insurance and safe deposit	10 66
" gift of Sir Ben. Baker	48 20	Contingencies	50
Total receipts for year	$5,168 88	Equipment (furniture, etc.)	12 75
Cash on hand first of year	2 62	Library and book purchase	171 00
	$5,171 50	Total expenditures	$3,977 41
		Cash on hand to balance	1,194 09
			$5,171 50
Cash on hand first of year 1898–1899	$1,194 09		

All money due the Association at the end of this year has been paid, and all outstanding accounts against it are paid to date of this report. The accumulated cash balance is held in reserve to meet the interest charge on mortgages falling due January 1, 1899.

ASSETS AND LIABILITIES.

Assets.

Cash on hand as above	$1,194 09	
House and lot, 12 W. Thirty-first street, New York City.	65,000 00	
Furniture and equipment	5,000 00	
Books and manuscripts	10,700 00	
Total assets	$81,894 09	$81,894 09

Liabilities.

First mortgage held by N. Y. A. of M.	$33,000 00	
Second mortgage bonds held by members of A. S. M. E.	8,400 00	
Second mortgage bonds held by council of A. S. M.E. as an investment	23,600 00	
Total liabilities	$65,000 00	$65,000 00
Excess of assets over liabilities		$16,894 09

These reports called for no action and were ordered printed as part of the record of the meeting.

The Secretary presented to the meeting the amendment to the Rules, of which due notice had been given as required, at Niagara Falls. The proposed amendments were as follows :

Article 10 is to be supplemented by the addition at its close of the following:

"Application for membership from engineers who are not resident in North America, and who may be so situated as not to be personally known to five members of the Society, as required in the foregoing paragraph, may be recommended for ballot by five members of the Council, after sufficient evidence has been secured which shall show that in their opinion the applicant is worthy of admission to the grade which he seeks."

And the sentence in that same article which now reads, "He must refer to at least five members or associates, personally known to him," is to be changed so as to read, "He must refer to at least five members or associates to whom he is personally known." The final sentence of the present article is to be similarly changed, and the words "personally known to him" are to be changed to "to whom he is personally known."

After some discussion looking to more careful limitations of the territory to be included in that part of the amendment affecting members not resident of the United States, the amendment proposed was amended, on motion of Mr. Jesse M. Smith, by substituting for the words "North America," the words, "United States and Canada." This proposed amendment having been passed, the motion came up on the amendment as amended, and was carried. The second amendment was then presented, involving the verbal change in Article 10, and on motion was carried.

The reports from professional committees being then in order, the reporter for the Society's Committee on Uniform Methods of Testing Materials, Mr. Gus. C. Henning, presented its report, as follows:

Mr. Gustavus C. Henning.—In the first place, as announced in the Report of the Council, I should like to state that the American Society of Mechanical Engineers has now assumed membership in the International Association for Testing Materials. I would like also to preface this report by the statement that it is mainly due to the active interest taken by the American Society of Mechanical Engineers that the International Association for Testing Materials has now taken a firm root in the United States; that an American section has been formed, and that, as was to be expected, all those specially interested in testing materials—college professors and others—have actually taken part and assumed membership, as well as some of our largest steel works, such as the Bethlehem Iron Works, the Carnegie Iron and Steel Works, and the Pennsylvania Steel Company; and we understand that at a very early date some other large steel works will also join. The work of the Association, as I have explained before, is entirely of a practical character. There are no abstruse questions involved. The work

to be done is strictly in conformity with the work of the engineer and the relation between the engineer and the manufacturer, in order to facilitate means and methods for obtaining such materials with certainty as are required and as can be furnished by the manufacturers, under the varying conditions existing in various countries or in different parts of the same country. I would like to emphasize this, because it has been mentioned by several who are not well posted that an association of this sort can lead to no practical results because it is composed of theorists. I would like to say now that the membership of the American section is composed, in the large majority, of manufacturers and men in business who have no interest in the abstruse investigations of the scientist pure and simple, but that the various manufacturers have come together to explain the possibilities of their processes and business methods, so as to let the engineers understand fully what can be expected of them; and then, through the action and the discussions of this American section, there will be a small variety of specifications and contract requirements so as to lead to economies and better understanding, facilitating the execution of contracts in every case where manufactured materials are used.

On behalf of the Committee, I desire to present the following:

Mr. Keep has constructed three autograph recording machines, to be used in connection with the work of the Committee:

1. An autograph recording transverse testing machine.

2. An autograph recording drill press for hardness determinations.

3. An autograph recording impact machine, having capacity up to 1 by 24-inch bars.

Tool steel bars of the following dimensions have been prepared to be tested by the latter: 6, 12, and 24-inch lengths; $\frac{1}{2}$ inch square; $\frac{1}{2}$ by 1 inch; $\frac{3}{4}$ inch square, and 1 inch square; the $\frac{1}{2}$ by 1-inch bars to be tested flat and on edge.

The hammers for this machine are of 25, 50, 75, and 100 pounds weight.

For lack of time, the cooling curves and temperature determination have not yet been made, but these and Mr. Keep's new work will probably be presented at the next meeting.*

GUS. C. HENNING,
Recorder for the Committee.

* I would, say, however, that Svedelius, a Swedish engineer, has investigated this same subject with use of the Le Chatelier pyrometer, using bars four inches

The second committee report was that on a Revision of the Code for Conducting Boiler Tests, which was presented on behalf of the Committee by Mr. Wm. Kent. Dr. Charles E. Emery, who had been acting as Chairman of the Committee, had passed away since the last meeting of the Society.

Mr. Kent.—The Committee presents the results of its labors in this pamphlet, copies of which have been distributed. At the annual meeting a year ago the Committee presented what was called the preliminary draft of the report, for the purpose of receiving criticisms. That preliminary draft was discussed in two meetings of the Society, and several written communications were sent in, and the discussion was printed for the benefit of the Committee. All the criticisms were carefully considered, and the result is this pamphlet before you, which is still subject to such revision as may be undertaken by the Committee before the date when it has to be handed in for insertion in the volume of the *Transactions*. There are some blanks left in the appendix, which will be filled in in the final pamphlet, and there are some portions of the printed discussion of last year that may be included in this appendix. Many of the points in the discussion of last year referred to matters which were in the preliminary draft of the report, and which are not in the present draft. That is, the criticisms were considered sound, and the objectionable parts of the preliminary draft were stricken out. So there will be no necessity of publishing these criticisms, since the thing criticised has disappeared. But other parts of the discussion, in which useful information has been given, may be put in the appendix.

Mr. Rockwood.—As I had not seen this paper before, and I assume that most of the other members have not, I move that this be not considered until the next semi-annnal meeting, for action.

Mr. Jesse M. Smith.—I do not quite see the necessity of having a standard method and an alternate method of conducting boiler

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long. He used the Fagersta steel, and found the very same behavior and the same cooling curves that we found in cast iron, only modified as affected by the composition of the materials. He has also identified the peculiar points in the cooling curves, with the points of recalescence, as determined by Osmond and a great many other investigators. So I think that when we finish our cast-iron investigation in the same direction, that we will then have a complete series of the changes that occur in bars of iron, using the word "iron" as a generic term, and at the some time determining the cause of such peculiarities.

tests. It seems to me there ought to be one or the other, with certain qualifications where special boilers or furnaces are being tested. I have just had a discussion with some parties about whether a boiler test shall be made by the standard method or by the alternate method. It seems to me that the standard method, where the fire is withdrawn entirely from the furnace just before beginning the test, and again withdrawn at the end of the test, is one which is not to be recommended. It seems to me that the "running test," or that which is known as the alternate method, is the better of the two, and it also seems to be the conclusion of the committee, judging from the foot-note applied to Section 8; or at least a great many engineers consider that the "running test" is the more reliable of the two.

Another point in regard to paragraph 15, "Treatment of Ashes and Refuse." "The ashes and refuse are to be weighed in a dry state." I think "ashes and refuse" should be defined more exactly. Certainly in a great many furnaces, and particularly mechanical stokers, there is a great deal of fuel that goes directly through the grates without hardly being charred. What is to be done with that fuel?

Mr. Kent.—The Committee will be glad to receive any suggestions from any members in regard to any further amendments that may be made of this report, and we will take it into consideration, and if the Committee agree to make the amendments, then they will make them and submit the final pamphlet for publication in the *Transactions*.

The Secretary.—At the spring meeting?

Mr. Kent.—It is now listed for Volume XX. We want to keep it open as long as it is convenient to you—that is all.

In regard to the particular case mentioned by Mr. Smith, I would say that we have carefully considered all the arguments in regard to the standard and the alternate code. I would call his attention to the foot-note on page 8: "Many engineers prefer the alternate to standard method on account of its being less liable to error due to cooling of the boiler at the beginning and end of a test." By putting both methods in the code, any engineer is at liberty to choose the one he likes best, or insert it in his contract. In regard to the treatment of ashes and refuse, it is generally understood that "ashes and refuse" is what is withdrawn from the boiler furnace. The ash all contains some coal, and the coal partially burned contains some ash, and

the thing needs no other definition. If any one wishes a finer distinction, he can make it in his own professional work. We do not think that paragraph should be changed. At the same time we are ready to receive any suggestions for change.

I would say that the amount of labor spent on this report has been enormous. Before Dr. Emery died, he spent several days going over every criticism and suggestion. Mr. Barrus and Dr. Emery and I met together and went over them, and every suggestion brought up has been carefully considered and this is finally the result of our combined efforts, with the written approval of all the members of the Committee except two who have not objected, but they have not yet signified their acceptance.

Professor Hutton.—Mr. President, if I may be allowed to make a suggestion which you have just made in my ear, it is that the Committee should consider the same suggestion that was made at Niagara Falls in respect to the standard method of testing engines—that there should be incorporated into the report a series of blanks, in a convenient pocketbook form, which could be disposed of by the Society on demand for a limited price to those who would like to use them. I think it has hindered, to some extent, the convenient use of our code that the only way in which our standard blanks could be secured was by buying a whole copy of the report for which many of the engineers did not care. It involved transcribing of headings, etc. in making the report.

The further suggestion of the President and one which I would like to commend in his name to the Society, is that the report should be made with blanks convenient for the metric system units as well as the American units. This suggestion also comes to us from English and continental sources—that our report would be international if we make that slight addition to it. They are ready to make use of it in Germany and France if that addition is made.

The Secretary in his own name would reply to the suggestion of Mr. Kent, that Volume XX. of the *Transactions* includes the *Transactions* of this meeting and the *Transactions* of the spring meeting. In that view it might be possible to carry out the suggestion made by Mr. Rockwood that the consideration of this report and the action upon it should be made an order of business at the approaching spring meeting. It would then come into the *Transactions* of this volume, and under that we need not decide on the action just now.

Mr. Kent.—There is no objection to that at all.

It was therefore understood that copies of the revised report of the Committee would be distributed in advance of the spring meeting of the Society, with a view to action upon the report at that meeting, if final agreement could be secured.

The next report was one of progress on behalf of the Committee appointed at the Niagara Falls meeting to prepare and submit a code for the uniform testing of engines.

Professor Jacobus, as Secretary of the Committee, then made a verbal report of the work to be undertaken and of the points discussed at the first meeting of the Committee. He stated that the Committee had taken no action and would not do so before communicating with the foreign representative, Mr. Bryan Donkin. This being the case the Committee thought it best not to publish the verbal report at this point, but to withhold all matter until a preliminary report could be agreed on by the Committee and presented to the Society for discussion.

Mr. H. H. Suplee.—I should like to ask if this report is intended to be used in conjunction with the work of the Thermal Efficiency Committee of the Institution of Civil Engineers, or is it entirely independent of it? They are working hard on this subject, and I believe Mr. Donkin is a member of that committee also. I think it was suggested at Niagara Falls that possibly this should be made an international matter. I would like to ask if anything has been done toward uniting the work of these two committees so that their reports will harmonize, or whether they are working entirely independent of each other.

It was finally suggested by Mr. Jesse M. Smith, that the text of the preliminary report should be distributed in advance of the spring meeting to all members, that they might have an opportunity of making suggestions to the Committee before the Committee took final action in making recommendations.

The President then called for the report of the Tellers of Election who had been appointed to count the letter ballots cast for officers of the Society.

The report of the Tellers was as follows:

REPORT OF TELLERS OF ELECTION FOR OFFICERS.

Your committee appointed to count ballots cast for officers of the American Society of Mechanical Engineers for the year 1898–9 begs to submit the following report:

Total ballots cast	564
Ballots thrown out on account of irregularities	32
Total votes counted	532

of this latter number,

530	votes	were	cast	for	George W. Melville	for President.
2	"	"	"	"	S. T. Wellman	" "
532	"	"	"	"	Wm. H. Wiley	" Treasurer.
532	"	"	"	"	E. D. Meier	" Vice-President.
529	"	"	"	"	George R. Stetson	" " "
533	"	"	"	"	B. H. Warren	" " "
529	"	"	"	"	Edgar C. Felton	" Manager.
534	"	"	"	"	R. H. Soule	" "
531	"	"	"	"	A. M. Goodale	" "

Respectfully submitted,

WM. J. BONNER,
JOHN H. BARR, } *Tellers of Election.*

The President.—If the chair hears no objection to this report I will officially declare the candidates named in the report elected to their respective offices in the Society, their duties to begin, under the constitution, with the adjournment of the present convention. I will appoint an escort committee, consisting of ex-President Warner and ex-President Billings to conduct the President-elect to the platform.

Messrs. Warner and Billings escorted George W. Melville, President-elect to the platform, and presented him to the President in the chair.

The President.—I have the pleasure of presenting to the American Society of Mechanical Engineers, its President-elect, George W. Melville, Engineer-in-Chief of the United States Navy.

Mr. Melville, on receiving the handshake of the President, turned to the Society, and thanked them in the following terms:

" Mr. President and fellow engineers:—The announcement that has just been made of my election as President of our Society for the coming year, notifies me of an honor conferred upon me which I appreciate beyond all others that have fallen to my lot. You can readily understand, gentlemen, that in our Naval Engineer Corps seniority or accident may sometimes effect the choice of the Engineer-in-Chief, but the position of President of our Society is one of which any man may well be proud, because it comes only as the result of your free choice. When I think of

the high order of talent of all our members, and of the great number of men of the highest professional eminence on whom your choice might have fallen, many of whom would doubtless fill the chair with greater efficiency than I can hope for, the honor you have done me seems all the greater, and I trust you will understand how thoroughly I appreciate it. In accepting the office, therefore, I beg you to believe that it will be a pleasure, as well as a duty, to do all in my power to prove worthy of your choice, and I shall devote myself in every way to the advancement of the interests of our Society."

Messrs. Henning and Ashworth, on behalf of the membership, spoke in fitting terms of the achievements of the President, and of the mutual honor that was conferred by the choice of their chief executive.

The President then announced that new business was in order.

Mr. Kent rose, and made the suggestion that the Rules of the Society, with respect to the appointment of the Publication Committee and which state its duties, should be revised by the Council. His suggestion was that it should be referred to the Council to consider these Rules, and whether the Publication Committee should not consist of seven members instead of five, and that Article 29 should be amended by adding a clause to the effect that no paper shall be accepted for final publication in the volume of *Transactions* until it has been approved in writing as being worthy of a permanent place in the *Transactions*, by three members of the Committee, the Secretary not to be included in those three. Mr. Kent's desire was that by such amendment, at least three members of the seven should have every paper prominently brought before them before it is accepted for final publication. He hoped that in this way the *Transactions* might be purged of what he considered bad engineering literature, such as is sometimes presented in pamphlet form at the open meetings.

Messrs. Wiley, Henning, Parsons, and Rockwood, spoke in explanation of the present methods and policy of the Publication Committee.

On motion, the general subject presented by Mr. Kent was referred to the Council.

The advisability of arranging for meetings of individual members of the Society during the interval between the annual meeting in the winter and the spring meeting six months thereafter, was presented by Mr. Halsey, as follows:

Mr. F. A. Halsey.—There is a considerable feeling in the Society to the effect that there is quite a reservoir of talent available to the Society that is not drawn upon for the benefit of the *Transactions*, namely, the younger element of the Society. The feeling is that, largely due to timidity, an element able to supply good papers holds itself back and does not do it, a situation which has been intensified through one or two unfortunate circumstances—papers from this source having been unduly "sat down upon," in several cases in discussion, and there is quite a feeling that something ought to be done to counteract this situation and to encourage the writing of papers by this element. The feeling is that there are a good many of the younger element who are capable of making good investigations and presenting them in creditable form. One plan that has been suggested is that special meetings should be held, to be called Junior meetings, to be devoted to these younger members of the Society, the proceedings of which should be in their charge, and which might or might not be incorporated in the *Transactions*. I am not prepared to make a motion, but I know that there are others who feel the importance of the matter as I have outlined it, and if so disposed they can present a motion.

On motion of Mr. Baldwin, the subject of providing for such extra gatherings, to be under the management of the Junior membership, if this were possible, was referred to the Council for it to take such action in the matter as seemed desirable.

There being no other or new business presented from any of the members, the docket of professional papers was taken up, and the two papers on "Strength of Fly Wheel Rims," by Mr. A. K. Mansfield, and on the "Bursting of Small Cast Iron Fly Wheels," by Prof. Chas. H. Benjamin, were presented and read and discussed. Participants in the debate were Messrs. Jacobus, Suplee, McBride, Cary, Frith, Fritz, Henning, Webb, Jesse M. Smith, and Van Derhoef.

The usual luncheon was served in the lower room at the conclusion of the discussion, in order that members might not have to leave the house in the middle of the day, but might remain for social and other opportunities during the afternoon, before undertaking matters of business which it might be their personal desire to see to. A committee of New York members was on duty to see that members were introduced to each other. In the later afternoon, by arrangements perfected through Mr. J. W. Lieb,

Gen. Manager, the members were welcomed at the Duane Street Edison Lighting and Power Station.

In the evening the annual reception tendered by the New York members to their guests was held in the new ballroom and anterooms at Sherry's, Forty-fourth Street and Fifth Ave. The President, Mr. C. W. Hunt, with his wife, received the members, and after the Officers' Reception, supper, dancing, and other social pleasures completed the evening.

Third Session. Friday Morning, December 1st.

This session was devoted to professional papers. Those allotted were, "Theory of the Moment of Inertia," by Prof. C. V. Kerr; on "Certain Improvements in Steam Boilers," by Mr. W. B. Le Van; on the "Generation and Utilization of Steam by the Lykens Valley Coal Company," by R. Van A. Norris; on the "Valve Gear of the Willans Engine," by John Svenson; on "Methods of Testing Indicators," by D. S. Jacobus; on "Variation of Belt Tensions with Power Transmitted," by Prof. W. S. Aldrich. The paper of Mr. J. H. Vail on "Cooling Tower and Condenser Installation," postponed from the previous session, was also read at this meeting.

Participants in the debate on the above papers were Messrs. Kent, Kingsbury, Bryan, Dean, Walworth, Rockwood, Allison, Wheeler, Ashworth, Suplee, Snell, Meyer, Jesse M. Smith, Halsey, Bole, Barth, and Oberlin Smith.

The Secretary presented an invitation, on behalf of the Engineers' Club, that the members of the Society should be its guests at an exhibition of lantern slides taken from photographs during a hunting trip for large game in South Africa, together with other invitations and announcements of courtesies extended to the meeting.

After luncheon Mr. B. C. Batcheller, pursuant to invitation, presented an illustrated description of the pneumatic tube system in use in New York City for the distribution of mail packets from the General Post Office. The description of his apparatus was illustrated with lantern slides upon the screen, and was listened to with great interest.

At the close of Mr. Batcheller's lecture, Mr. R. W. Hunt, on request of the President, made himself the mouthpiece of a friendly conspiracy, whereby the friends of Mr. John Fritz had

secured from him the handsome three-quarter length oil portrait of himself, which he had been persuaded to present to the Society, but which it had been his desire to hang upon the walls of the auditorium without formality. The many friends of Mr. Fritz, felt, however, that it was more fitting that this pleasant event should be a public one also, so that the portrait had been secured and hung on the walls, properly veiled. At the conclusion of Mr. Hunt's genial remarks the veil was withdrawn and the members had the pleasure of seeing how successful the artist had been. It was left for the Council to take formal action in recognition of Mr. Fritz's gift to the Society.

The members, at the conclusion of this presentation ceremony, adjourned to the car house of the Metropolitan Traction Company, where special cars were assigned to the party for their transportation to the General Post Office. The Traction Company added to the courtesy of providing for the Society by carrying the party without charge for fare. After the visit to the Batcheller Pneumatic Tube System the party broke up into individual groups.

No assignment was made for Thursday evening in any official way, and many members attended the gathering at the Engineers' Club or amused themselves in their own preferred way.

Closing Session. Friday Morning, December 2d.

The professional papers of the morning were those by Messrs. R. S. Hale on the "Calorific Power of Weathered Coals"; by Mr. W. H. Bryan, on "A Mechanical Plant of a Modern Commercial Building"; and by Prof. R. C. Carpenter, "Experiments on the Flow of Steam Through Pipes." Besides this, a paper by Mr. Chas. H. Newcomb, detailing tests on flow of water through hydrants, was presented in manuscript, with a limited number of its illustrations.

The participants in debate were Messrs. Dean, Kent, Christie, Jesse M. Smith, Geo. Hill Bolton, George H. Hill (by invitation of a member), and William Kent.

At the close of the professional papers certain queries which had been submitted were presented for discussion. Messrs. Woolson, Newcomb, Suplee and Kent presented their views in reply to the query, "Does it Pay to Pickle Ordinary Castings"; Messrs. Fickinger, Souther, and Kent discussed the meaning of

the term, "Seamless Tube"; Messrs. Jackson, Kent, Jesse M. Smith, and Halsey gave their views on the question on "On how Small a Tool does it Pay to Put an Individual Electric Motor." Messrs. Newcomb, Cary, Durfee, Sanguinetti, and Dallett discussed the subject of "Machine Shop Floors."

This completed the regular order of business, and at its end the Secretary asked that a motion might be passed tendering to the Metropolitan Traction Company the thanks of the Society for the courtesies enjoyed at its hands on the occasion of their visit to the pneumatic dispatch tube system on Thursday afternoon.

The President expressed to the meeting his sense of appreciation for the kindnesses and consideration which had been shown him during his term, and handed the gavel to his successor, present with him on the platform at the hour of adjournment. The motion to adjourn having been put and passed, the meeting ended.

It may be added that by a vote of the Council at a meeting held after the adjournment of the general meeting, the city of Washington, D.C., was chosen as the place for the spring meeting of the Society next to follow.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS—ANNUAL ADDRESS OF THE PRESIDENT, 1898.

Forming a part of Volume XX. of the Transactions.

BY CHARLES WALLACE HUNT.

It has been the custom of this Society for the President to deliver a formal address at the end of his term of service. This practice, like other acts which continue long enough for the establishing of a custom, must have good reasons for its existence, although they may not be fully appreciated either by those who established or by those who follow the custom.

The duty which devolves upon me this evening was first entered into as a task, but it grew to be a pleasure, as the growth and importance of the functions of the engineer became evident on every side, as we study them in our national development, in our industries, and even in the comforts and the luxuries of our daily life.

Each one of us looks out upon the same world from a different standpoint, and each sees the same general scene, but the scope of the view and the details observed will vary to a greater or a less degree, depending upon our particular position. In addition to this, each, as it were, looks through a colored glass which gives a personal tint to the scene, colored by the effects of our environment, as well as by our personal temperament.

Could we combine all these various pictures, large and small, which are presented to our view, with all their varied tints, we would obtain a kind of composite image, which would be a more accurate and probably a more pleasing representation of the real subject than any one of us sees individually. A senior, who has travelled the rugged path of life, should be able from his experiences on the way to select such views as would be both useful and pleasant for a junior to consider as he starts out on a similar journey. The interest of our annual meeting is heightened, and an intellectual pleasure is given us, when one of our body presents to the others those subjects which seem to

him important and interesting, that all may compare them with the view as seen from their own standpoint. As we pass from subject to subject, each will combine the picture presented with his own personal conceptions, giving, as it were, a stereoscopic effect, each one gaining a wider and a clearer view. In making this survey we will first consider those matters which immediately concern engineering practice, and then pass on to wider fields and subjects of more general interest.

The Word "Engineer."

In order that we may proceed in harmony of thought, we must use words in the same sense. Let us, then, first consider what we mean by the word "engineer"—not what it meant historically, but what it has come to signify in the active world of to-day—and try to bring our individual conceptions of this meaning into harmony each with the other. Following Tredgold, I have herein used the word "engineer" in the broad sense of one who is skilled in the application of the materials and forces of nature to the uses of man.

Considered in this broad sense, the engineer is interested in every investigation and discovery in the whole realm of nature. Experience has shown that every field is tributary to his work. The theoretical abstraction of yesterday becomes a demonstration to-day, and to-morrow it is the task of the engineer to apply it to the uses of man. The new discoveries of materials, of forces, and of laws which now succeed each other so rapidly, make a corresponding increase in the range of the work and the responsibility of the engineer of this present day.

Engineering Practice.

That we live in an age of changes is at once our opportunity and our pleasure. Some of these changes burst upon us, attracting universal notice, while others come so slowly that they are almost unobserved. A change of the latter character has been taking place of late years in the work of professional engineers. This has largely come from the development of our manufacturing institutions from the position of being a minor factor in our economic life to being one of commanding importance, and the

necessary employment of skilled engineers to conduct their technical affairs.

The engineer of the user and the engineer of the maker have widely different duties. Consider how different may be the information required in practice by two classmates, whom we will designate as "A" and "B," who graduate from college as engineers. We will suppose that "A" secures a position in the engineering department of a city, and commences his work, which may be the designing of a new water-pumping station. His college course has fitted him for the work. His text-books were suited to problems of this character. He finds abundant information on all branches of the subject, in data published in the proceedings of scientific societies, in technical literature, and in annual reports of city departments. The forms of contracts to be entered into are at hand, all found elaborately drawn, with every point safe-guarded, and need only a little selection and adaptation to suit his case. They place in his hands the power to decide absolutely and without appeal all questions which may arise in carrying out the work.

"B" obtains employment in the engineering department of a manufacturing corporation, which in due time is to submit a tender for the construction of the pumping plant for which "A" has issued specifications. He will find that the form of contract proposed by "A" has many minute and carefully worded clauses to bind and limit the supplier. The tender to be submitted for the execution of the work must in its scope and wording protect the interests which "B" represents, not only in a general sense, but in every one of the clauses of the proposed contract. Every obscure phrase and every adjective used by "A" must have definite consideration and be clearly defined in both an engineering and a legal sense. "B" here finds that the information derived from his college course is meagre, and there is no technical literature which he can use, either as a general guide for making a form of tender, or the proper expressions to use to define or limit the obscure clauses or words found in the specification.

Looking at the subject from a purely technical point of view, we see quite as great a variation in their work. In the case supposed, "A" would require only a general knowledge, while "B" would require the most thorough and exhaustive information of the qualities of constructive materials, and shop practice

available in that particular location. The farther we carry the comparison of their work, the more clearly it is seen that the educational needs are becoming more and more complex, to correspond with the growing specialization of engineering work.

Practice Abroad.

There is another phase of engineering practice represented by the duties of "A" and "B" which now becomes interesting, if the work of American engineers is to take the place in the world at large to which the indications now so plainly point. In other countries it is a common practice for "A" to make all the general designs and all of the details for engineering work, and the supplier has no responsibility for either, or for the efficient working of the plant when completed. If errors or omissions are found in the drawings or specifications, the cost of the changes required is paid by the purchaser, in the usual bill for extra work. In this case, the duties of "A" are exhaustive, and those of "B" are small or disappear altogether.

American Practice.

The American practice is tending to the method of making the requirements issued by "A" of a general character which will cover the results sought, and leave to the supplier, "B," the work of designing the particular means to accomplish the desired end. Business has become of such a magnitude and so complex that one mind cannot fully grasp and readily handle the new discoveries, new materials, and new practices which now come so rapidly. For efficient and economical results, each phase must be handled by an expert.

There will be many "B" engineers to respond to the requirements of "A," and each will present for consideration different ideas, different materials, and different shop practices. "A" must select, from these various plans and details submitted, the one which best promises to fulfil the requirements. It is a division of labor between "A" and "B," each of whom, by tastes and training, is especially fitted for his part of the work. We may paraphrase their duties by saying that "A" is a judge, "B" is a counsellor.

Post-Graduate Work.

At the present time we cannot expect our technical schools, painstaking and perfect as they are, to fully prepare both "A" and "B" for such new and varied duties, or even to have their instruction in engineering fully abreast with the latest practice, or at least not until progress in the arts and sciences has substantially ceased. It takes time for a new practice or a new result to be recorded, published, considered, and adopted by the teaching staff.

This difference between the teaching and the engineering practice of the day is not only an indication of progress in engineering, but in some measure is an index of its rate. The student, then, must expect, as a normal proceeding, to supplement his graduating acquirements by practical experience, together with a personal contact with his professional brethren, in order to place himself fully abreast of the times, and to be fitted for the most effective and useful engineering service.

Engineering theory and practice are rapidly extending with the general advancement of our economic interests, and the engineer, whether he be a young graduate or otherwise, who does not make use of the modern aids to information, among which are to be counted scientific societies, and a personal association with his brethren, with the innumerable hints and suggestions which come from these, will soon be found struggling with what seems to him adverse fate, but what, in reality, is inferior knowledge, behindhand knowledge, or, plainly speaking, ignorance greater or less. The engineering world has passed by him, and he must then view the working out of the law of the survival of the fittest with what grace he may.

Laboratory Development.

An interesting development in the engineering world of the present day is the rapid growth of the experimental equipment of our colleges and technical schools. There seems to be no limit to the expense and the completeness of the illustrative and experimental machinery which is being installed for the instruction of the students of these institutions. And not less valuable is the learning, industry, and skill of the professors

in charge of and directing these schools, whose theoretical acquirements are supplemented by being in constant personal touch with the industrial and economic interests of the country.

It is possible that by an organized effort the magnificent equipment of trained professors and experimental apparatus could be brought in closer touch with each other, that to a material extent their work and investigations might be made to proceed on a predetermined plan. This would broaden their field of experimental investigations, lessen the duplication of work, systematize the publication of results, and more rapidly extend our growing fund of accurate engineering data.

A Helping Hand.

The engineering and scientific work of to-day uses one or the other of two systems of metrology,—the English or the metric. The discussions of the relative importance and the desirability of these systems of weights and measures are frequently interesting, and may to some extent be useful in familiarizing the terms and making easier the conversion of quantities from one system to the other. Practical engineers can, however, lay aside academic discussions on the advantages or disadvantages of either, and recognize that the two great systems of metrology are each used by great engineering nations to the practical exclusion of any other, and they may safely assume, without discussion, that they are not likely in the near future to be changed in any material way by those using them.

It is especially desirable that English-speaking societies shall give every practicable aid to engineers using the metric system of measures, that the work of their engineers may be readily available and with the least possible trouble in making conversions of quantities from the English to the metric system. Such computations are always troublesome to perform, and distracting to the mind when undivided attention is required by the subject matter of the article. If the numerical expression in English measures is followed in a parenthesis by the exact metric equivalent, the article is practically translated when printed, as most engineers using the metric system read the English language, although they may not speak it, or readily make numerical conversions. The greater the availability and

the publicity given to the published proceedings of a scientific society, the more nearly has the society accomplished the chief object of its existence.

An Extending Field.

It has long been evident that we were making rapid progress in perfecting our manufacturing machinery, as well as organizing and developing our industries, thus constantly increasing the efficiency of our labor, until we have reached a point where an hour's labor with its facilities produces more of our principal products, and transports them farther, than an hour's labor will do in any other part of the world.

The late war has revealed to us the fact that we have gone on reducing the cost of our products, and increasing our capacity for production, until our country alone does not furnish a sufficient market to insure steady work for our labor, and prosperity for our merchants and manufacturers. Like confined waters, the tendency of these increasing economic forces has been to break out from their confinement and equalize trade conditions by seeking a market in the world outside. If articles which are necessary to supply the wants of man can actually be made here with less labor and cheaper than elsewhere, here they will surely be made, though it modify our traditional ideas of isolated position, and our protective theories.

One hundred years ago the iron trade of Sweden was greater than that of England, and remembering the great changes which have taken place in the last hundred years, it would be rash to assume that the momentous economic changes which are now taking place, may not cause an equally great shifting of the centres of more than one phase of industrial activity.

Scientific Societies.

Every age has produced most ingenious and able engineers and mechanicians, as is conclusively shown by specimens of their work. Many of these have been preserved and handed down to us, causing us to wonder at their skill when we consider the limitation of materials then available, and in their time the paucity of exact knowledge of the laws of nature. But the special knowledge and experience of those masters in the art

practically disappeared when death claimed the originators, as only a small portion usually remained in the minds of the pupils, and but little of this was transmitted to posterity. It was only when scientific and technical societies for the preservation of accurate records had developed, in the fulness of time, making all the world pupils, that the valuable knowledge so laboriously obtained was preserved and handed down to those who, sooner or later, could utilize it for the comfort and the advancement of mankind.

These lately developed scientific societies, which are so prominent a feature of the present age, were organized for the discovery and the universal diffusion of scientific knowledge, an object entirely different from that of all mediæval guilds and trade organizations. At first they were largely philosophical, discussing theories and experiments which at the time appeared to the community at large to have little or no direct bearing on the practical affairs of life.

The members presented to the society the results of their investigations and experiments, in the form of written papers, making them permanent records to be consulted and made available by others who were contemporaneous or who would succeed them. This was the vital germ which was to develop and elevate science and its applications in industrial work in succeeding ages. These societies thus became, so to say, the savings-banks of our civilization, the repositories and guardians of the results of investigations, experiments, and experience that otherwise would have been lost to the world.

As industrial interests became more important, other societies sprang up, each devoted to some particular phase of scientific or technical work; each gathering, selecting, and recording data, not alone for their members, but to become permanent additions to the general fund of scientific and engineering knowledge. The growth of these societies has been accompanied by a gradual decrease of secret methods of manufacture, formerly so prominent, but which have now practically disappeared in our industries. Manufacturing supremacy is now decided by other factors.

The advance made in the accumulation of useful data and more accurate knowledge in practical engineering gained one season, is presented to a scientific society the next, and still later it will be embodied in text-books for the instruction of students who are soon to take our places and carry on our work.

Until attention is called to the subject, we are not likely to realize that, in their essential parts, the great bulk of the engineering data available to us now has been first presented to a scientific society, and there permanently preserved until the time came for its utilization or application. It is this great fund of information, principally accumulated during the last century, that we draw upon for the materials for our text-books, our general treatises, and our engineering hand-books.

Applied Science.

Turning now to the effects of this accumulation of scientific data and literature available to all alike, and the results following its application by organized methods of procedure, our first glance will show prominently the wonderful and rapid increase of the importance of engineering in our industrial life. It has transformed almost every phase of it, and put into our hands materials and processes which make the actual life of our immediate ancestors seem primitive by comparison.

Commencing under adverse conditions and developing in a field of restricted capital, with scarce and high-priced labor, engineering in America has applied the forces and materials of nature to the uses of man in a characteristic way. Freedom from mediæval traditions and the hampering conditions found in the older countries left them substantially free in the choice of means to accomplish their end. Influenced as our engineering has been by the experience and the work of other parts of the world, yet we cannot escape the fact that its development was essentially independent, and in some phases unique.

Improvement has followed improvement in technical matters, profits and savings have been added to the capital invested in our industries, until our country, two hundred years ago an untraversed wilderness, one hundred years ago a struggling nation—struggling with industrial difficulties and serious political problems—has triumphed over those early limitations, and has developed into a nation which in numbers, prosperity, and wealth takes a prominent position among the great nations of the world.

The Advent of the Engineer.

Whichever way we turn, we behold marvellous changes, which have followed the advent of the engineer on the scene. A view

of one subject will in a measure serve to represent these changes, and to recall similar illustrations to your minds which differ from this only in degree.

It is but a few years, well within the memory of men now living, that our navy and all the other navies of the world were composed of sailing ships. In one of these vessels a mechanical germ was introduced in the form of a steam engine and an engineer. The grave question soon arose as to what should be the status of the new intrusion into the personnel of the ship, the engineer. This factor, which was soon to revolutionize the navy, was considered unimportant at that time, as is shown by the first official record on this subject in the Navy Department at Washington, stating that it would seem that such persons should be exempt from the penalty of corporal punishment.

The engine grew in size with each succeeding vessel, and as it increased, the sails correspondingly shrank, until finally they disappeared altogether. Other mechanical germs also found a lodgment in the ship, which have so developed that hydraulic and pneumatic pressures are produced, and electric currents are generated and distributed, to govern the rudder, hoist the anchor, ventilate the compartments, energize the combustion, revolve the turrets, train and control the guns, handle the ammunition, and purge the ocean's water of its impurities, making it wholesome for the ship's use.

Following these in quick succession came incandescent lamps and search-lights, breech-loading and rapid-fire guns, multi-charge automatic guns, and mobile torpedoes—one mechanical appliance rapidly following another, until the ship-of-the-line, which but just now embodied the result of hundreds of years of thought and experiment, has been completely transformed from keel to topmast into a vast machine, controlled and operated, even to the least important function, not by sailors, but by mechanicians.

In every phase of our industrial life the changes wrought by the engineer are quite as evident; for instance, note the marvellous changes in the manufacture of steel,—in the development of electric locomotion,—in iron building construction,—in machine tools,—in agricultural implements,—in sewing-machines,—in textile industries,—in electric metallurgy.

His Work.

The life of the engineer has a full measure of the labors, the trials, the discomforts, and the disappointments which are found in this as in every other walk of life. But it also has the successes which come from well-directed labors. It is not, however, either the useful work in itself, or what are called the successes of life, which brings happiness. It is man's ideals which make him happy.

Let us together survey some of the surrounding influences which tend to give high ideals of life to the engineer, no matter what the trials or the vexations of the moment may be. We will pass in review the interesting character of his daily work, his pure-minded associates, the higher pleasures of life, and the fascinating scenes by which he is surrounded. We will then better appreciate with what elevated emotions a father can lead a son, or a teacher his pupil, to the path of an engineering life, and place in his hands the mathematical, chemical, and physical implements to enter upon a work which will bring to him usefulness, pleasure, and honor.

Whichever way engineering may develop as time rolls on, its elevating influences are constantly at work on the mind and on the character. The work is carried on under unchangeable laws, which must be rigorously applied and adhered to, or failure is sure to result. Man builds to master, to resist, or to guide the forces of nature. If he has rightly judged the conditions, his work stands as a permanent monument of the fact; but if otherwise, the irresistible laws of nature will develop the defect and discover his ignorance, incompetence, or error to every observer.

His Researches.

Hence he laboriously seeks out the unseen laws and forces of the universe, then expresses the revelation in a workable form for his daily use. He tests his materials with painstaking refinement. He measures electric resistances with an accuracy now reaching the point of one in four millions,—time to the one-millionth part of a second;—divides a circle with a mean error not exceeding the one-millionth part of the circumference;—makes surfaces six inches square with a variation from absolute flatness of less than one two-hundred-thousandth of an inch,

and parallel within one second of arc;—rules lines which vary from absolutely perfect spacing by only one three-millionth part of an inch;—measures his optical work with a wave-length of light as a unit of distance, and handles this unit of the one forty-thousandth of an inch as easily as a mechanic handles a rule;—sees clearly the spectrum of samarium when one part is diluted with three million parts of lime;—and surveys lines eleven miles long, in the open air, with an average variation in three measurements of only four-tenths of an inch.

His Ethics.

The effect of living and working in such a sphere of action, where it is inconceivable that an engineer could knowingly be otherwise than exact in his work, should tend to influence the whole trend of his life and character, and make them to a greater or less degree a reflex of his daily work. He of all men has the most unchangeable and exalted basis for his ethics—the clearest of all knowledge of the disastrous results which will surely follow the violation of law. The very qualities of his mind which make his work a pleasure and a success will all tend to bring his every act into compliance with the inexorable laws of the universe. If it is otherwise, and his conduct is not guided by, and his ethics are not in accordance with the laws of right doing and right thinking, then, and to that extent, he is not an engineer—not one who is skilled in the application of the laws and the forces of nature to the uses of man.

His Pleasures.

It is with hesitation that I ask you to contemplate the pleasures of life enjoyed by those whose daily walk is thus surrounded. Words fail to describe the exquisite pleasures and the noble aims which are inspired by the contemplation of the wisdom and beneficence of the laws of the universe, which the diligence of man has revealed to us. Who can estimate the satisfaction which comes to the mind of the engineer from the knowledge that his work, the fruit of his investigations, and the wisdom of his decisions will be judged, not by fallible human methods and its caprices, but by the infallible and immutable laws of the universe!

Then consider the pleasure which comes from working in the open air, in the broadest light, where every interested one can see his difficulties, his investigations, his adaptations, and finally, if God has given him ability equal to the task, his solution of the problem. When victory comes, he is given the honor due to the work in unstinted measure, and he can accept it with propriety and count the commendation as one of the pleasures of life.

It is inspiring to the earnest engineer to feel that the actual workings of his mind, and his inner and fundamental conception of the forces of nature, of resistances, of materials, of workmanship, will be shown in his works as in a mirror. Roebling, Ericsson, Sir Benjamin Baker, and Edison have worked, as it were, in a glass house. Their thoughts and judgments are shown to all the world, not by inadequate words, but in the works of their hands—the Brooklyn Suspension Bridge,—the turreted Monitor,—the Forth Bridge,—the quadruplex telegraph,—the enclosed filament whose electric conductivity increases with the current.

Then he has the gratification to the mind which is found in comprehending and intellectually seeing, as clearly as in a diagram, the theoretical lines of the forces in a structure, and then clothing those lines with materials of strength and resistance, to make them realities, and adapted to do the everyday work of life. The Brooklyn Suspension Bridge by Roebling shows an almost ideal correspondence of the two, so that it may represent either theory or practice, depending on which way at the moment he chooses to look at it. Again, he uses a system of weights that cannot be seen or handled, the purely intellectual atomic weights—yet the rock under our feet is not more firm and real than is the work done with these intellectual aids.

His Environment.

Working in a field and in touch with a body of his fellow-men having similar tastes, he sees on every hand scenes of engrossing interest—the telescope photographically recording the position and motion of stars which no human eye has ever seen,—the spectroscope analyzing the materials of the sun and stars with all the accuracy which it would show if the articles were in the laboratory,—looking with Roentgen rays through a double-

barrelled rifle, and seeing not only the leaden bullets within the steel barrels, but also the wads and the charges,—and photographing lines in the ultra-violet and infra-red spectrum far beyond the reach of our vision.

He stands by a quartz filament galvanometer which indicates an electric current so minute that if it should be increased in magnitude eight hundred thousand times it would still be only the one-millionth part of an ampere; and on the other hand, in contrast, sees the Niagara electric generator of five thousand horse-power, with a current so much larger than that of the galvanometer that the difference can only be expressed mathematically, not in colloquial language. He sees with entrancing interest the liquefaction of hydrogen at a temperature of only twenty-three degrees centigrade above absolute zero,—and, again in contrast, sees what promises to be a rosetta-stone in astral analysis, in the precise correspondence of the spectrum of the star gamma Cygni and that of the chromosphere of the sun.

He shares in the enthusiasm at the results of two years of unremitting work in the extreme part of the known spectrum in isolating a new element, monium,—in the Hertz electro-magnetic waves now applied in wireless telegraphy,—in the newly discovered element polonium, whose radiations make the air through which they pass a conductor of electricity.

More nearly touching him personally comes the work of the biologist, whose quest for the thing we call life has continued from the primitive man to the present time. Constantly flitting from his grasp, it has seemingly passed from fire and storm to mountain and deep, from animal and plant to flower, to seed, to cell, and now it has been followed to the molecule or the atom; and yet it as completely eludes his grasp, or even his comprehension, as ever it has. But followed it certainly has been, by all the laws and forces of nature at the command of man, until the search for it is now in the atom, a space physically so small that only the trained imagination can even faintly comprehend its minuteness.

And there, on the outskirts of this unexplored world, stands man, with spectroscope and polarized light, peering into the sphere of action which we call an atom, well knowing that therein lie wonderful forces, activities, and at least the effects of that mysterious entity, life itself. He sees a field for investigation so fraught with possibilities, so infinitely beyond the com-

prehension of any conception which we can form of the capacities of the human mind, that he stands gazing into the abyss with the same devout wonder and awe as does the astronomer when viewing the illimitable heavens. The two are standing, as it were, back to back, and each is gazing into an infinity—one into the infinitely great, and the other into the infinitely small.

Thus stands the engineer in the midst of a countless number of earnest explorers in the field of unrevealed nature, and, so to speak, sees the tools forged and the materials discovered with which he is to work. Cheerfully can he enter upon his daily task with the consciousness that his application of these discoveries is of real service in lightening the burdens of our life, as well as elevating and ennobling his fellow-men.

The scenes which we have just brought to our intellectual vision are not those of the untrained imagination, of rhetoric, or of unreality, but those of the most rigorous truth, among the most real and matter-of-fact things known to us in all the realms of nature, and brought before you in the plainest language at my command.

We have traversed a wide field together, and now, as we draw near to a personal parting—never to meet again under similar circumstances—let us, as we travel the way of life, appreciate its elevating pleasures, and carry to our daily tasks and to our homes a higher realization of the dignity of the life and of the work of the engineer.

DCCXCIII.*

VALVE GEAR OF THE WILLANS ENGINE.

BY JOHN SVENSON, CHICAGO, ILL.

(Member of the Society.)

In the critical examination of any mechanism, simplicity is without doubt the highest test of excellence. When, therefore, in the gradual evolution of a certain piece of machinery great simplicity has been approached without loss to the result required, that particular combination of mechanical elements becomes an object of general interest. It is this thought which furnishes me with an apparently valid motive for leading you to a moment's notice of a steam-engine valve gear which has in its make-up so few moving parts, and which effects a steam distribution so admirable, that it may well be reckoned among the mechanisms above categorized.

The valve gear is not new of which I am about to speak. It is probably known to many of you as the most interesting feature of that engine which, of all strictly high-speed prime movers, has attracted the greatest attention in Europe, and which was the cause of bringing the late P. W. Willans before the world as one of the foremost steam-engine experts of our day.

While thus lacking in absolute novelty, my subject—so it has been suggested—may nevertheless interest those of you who have not as yet become entirely familiar with this admirable gear.

The valve mechanism of the Willans engine is so closely interwoven with the other reciprocating parts that, in order to render a description of it at all comprehensible, it will be necessary to include here a few brief references to the engine as a whole.

From a study of Fig. 1 we see an engine of the vertical type with single-acting cylinders—in this instance two in number—arranged tandem fashion over a common crank shaft. A peculiarity of design will strike you at once in examining the piston

* Presented at the New York meeting (November, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

rod. Contrary to usage, this element is here a tube, open at both ends and finished accurately inside and out. It runs steam tight through glands provided with elastic packing rings. At proper intervals it is pierced with openings arranged in transverse rows, through which the steam is made to enter and leave the cylinders, the admission and exhaust periods being regulated by the valve and by the movement of the rod itself. The valve

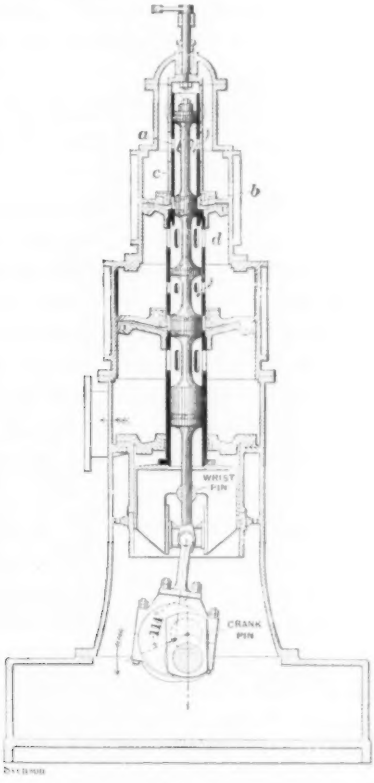


FIG. 1.

is a series of connected pistons which works inside the hollow piston rod. It is made to run steam tight by means of spring rings.

To the other portions of the valve gear proper, the briefest kind of reference will suffice. They are an ordinary eccentric with its strap; a rod secured to the strap in the usual way, and to the valve by means of the only pin about the whole gear; and, finally, a small cross head with guides.

As the piston rod is also the valve seat, the rod and the valve must move in relation to each other. This relative motion is had by mounting the eccentric upon the crank pin.

The valve opens the inlet ports; it also alone controls the exhaust, but it does not cut off the steam supply at the end of the admission period. That office is delegated to the care of the hollow piston rod, which then to that extent becomes a member of the valve gear. The manner in which the rod performs this part of its work will be best described in connection with the steam distribution. I will merely mention here that the arrangement not only permits a variable automatic cut-off, but it also gives rise to a rapidity of cut-off not heretofore attained with so few moving parts.

The eccentric has no part in varying the cut-off. This member is unalterably secured to the crank pin, and it has been found that an angular position of 114 degrees ahead of the crank is the most advantageous.

The steam distribution is briefly as follows:

When the parts are in the position shown by Fig. 1, the crank being at the upper centre, steam from the chest has free access into the hollow piston rod between the two highest valve spools. Ports *b* are also opened to the extent of a slight lead. As motion commences, these ports *b* are very rapidly opened. That such is the case is plain from the fact that the eccentric at the beginning of admission moves very nearly in the position of maximum speed of valve travel. On the downward stroke, while steam rushes in through the ports *a* and *b*, on top of the piston, a point will be reached when the ports *a* will be covered by the solid portion of the sleeve *c*. Then the inflow of steam will cease and expansion begin. Just before the crank pin has reached the lower centre, the valve, which is now on its upward path, has arrived at a point where the lower edge of the second valve spool coincides with the lower edges of the ports *b*. This marks the beginning of the exhaust period. The same rapidity of opening which characterized steam admission is again repeated in the release, as the valve now also moves in close proximity to the locality of greatest speed. The exit of steam is through the ports *b* and *d*, into the space beneath the piston, which space is either a receiver, or an exhaust chamber connected with the exhaust pipe, as the case may be.

I need hardly say that the cycle described above for one

cylinder is repeated in every other cylinder located beneath. If the gear is used on a simple engine the valve has three spools or pistons, and the operations noted end the matter. In a compound or triple-expansion engine two more spools must be added to the valve for every cylinder below the first. No matter how many cylinders may be used in one line, the same events of admission and release occur in them all simultaneously. The points of cut-off may, however, be varied to suit conditions.

I will return to the first portions of the cycle to consider for a moment the cut-off arrangement. It should be first noted, then, that when no variable cut-off is desired the sleeve *c* is removed, the upper portion of the hollow piston rod somewhat shortened, and the packing gland adjusted to the proper height, so as to serve the purpose of arresting the steam supply through the ports *a*, at the proper point.

The automatic cut-off device is a sleeve *c*, fitting nicely on the outside of the piston rod, and arranged to rotate when acted upon from the outside by a centrifugal governor. Reference to Fig. 2 will indicate the purpose and effect of such rotation. The inlet ports *a* of the down-moving piston rod, being in communication with the steam chest only through the triangular apertures cut into the sleeve, will become closed earlier or later, dependent on the angular position of said sleeve. By this means the period of expansion is varied to correspond with momentary changes of load.

Where the demand for accuracy of regulation warrants the expense, a similar cut-off sleeve may be fitted on the piston rod above the low-pressure cylinder also.

The indicator cards reproduced in Figs. 3, 4, and 5 will serve as a good text for a brief commentary upon the distribution attained with this valve gear. They were taken from a 240 indicated horse-power triple-expansion engine when running at a speed of 380 revolutions per minute. The engine was designed for condensing, but at the time of the test it exhausted into the atmosphere, and that fact will help to explain the loop in the low-pressure card. It is evident that the indicator springs are responsible for the irregularities of admission lines in Figs. 4 and 5, and of the exhaust line in Fig. 3. Fig. 3 is a good indicator of the valve action. The admission and exhaust lines are both excellent, the former dropping only 4 pounds towards cut-off. The quick cut-off is a point of no less interest. In fact, all

the periodical changes in the cylinders must have been marked with distinct and abrupt beginnings and ends, in order to result in the cards shown.

I have made a rough calculation of the time it takes to close

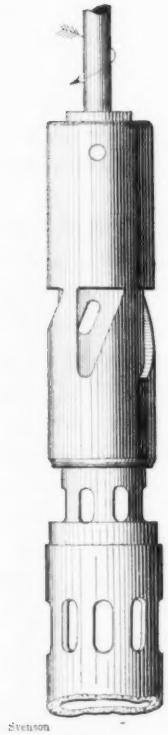


FIG. 2.

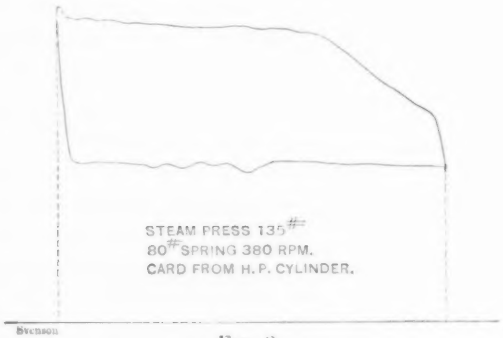


FIG. 3.

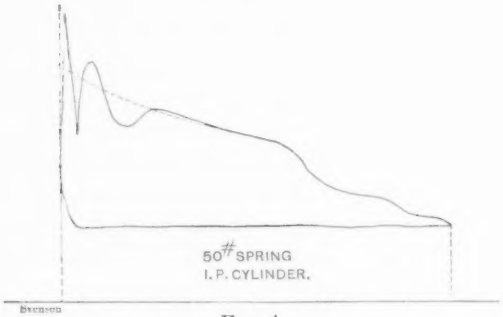


FIG. 4.

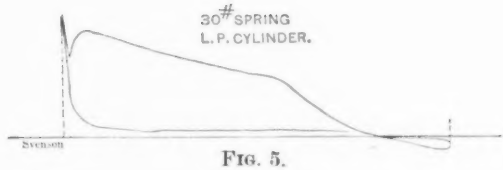


FIG. 5.

the inlet ports from full opening, and find that in the engine from which the indicator cards were taken it requires about $\frac{1}{50}$ part of a second. The rapidity is simply measured by the piston speed, and the time spent is equal to the time it takes the piston to travel the length of inlet openings (a).

Commenting further on the various characteristics of the gear, it is interesting to note that no matter how many cylinders may be in use, so long as they are placed in one line the valve gear still retains but one pin connection and one eccentric. The gear is, therefore, in the principal sense, no more complicated in a triple-expansion engine than it is in a simple engine. The valve, to be sure, must be longer, but an addition of valve spools is the least objectionable, as furnishing but increased weight. This feature of simplicity in a multiple-cylinder engine becomes of the utmost importance when an engine of high rotative speed is considered. The fewest number of jointed parts is here absolutely essential to maintenance, long life, and safety. A gear, then, which provides the desired simplicity enables the high-speed engine to multiply its cylinders, and will for that reason permit it to take its place among the economical engines of our time.

The locality of the valve and its operating mechanism in the very centre of the engine develops several very interesting advantages. The steam passages become singularly direct, clearance spaces small, and the general compactness is rather conducive than otherwise to the lessening of heat wastes. All strains, barring those due from the angularity of the connecting rods, are transmitted along the centre line of the engine, and the greater number of the construction details take naturally that form—the round—which is without question the most favorable for shop manipulation.

I need not enlarge on the evident fact that the valve friction is no greater than the amount due to the tension of packing rings. In this sense, then, the valve is unquestionably balanced. From another point of view it may at first sight appear that the valve suffers resistance to motion in overcoming the unbalanced steam pressure which is allowed to act with full force on the top spool. But while this pressure, of course, is a load on the up-stroke, it tends on the down-stroke to assist the crank, so that the total effect should be *nil*.

It would not be necessary to allow any top pressure at all when the gear is used in a slow-running engine. But in connection with the higher rotative speeds, the frequency of stress due to oft-arrested momentum of the valve mass would seriously tend to rupture, if not neutralized. The same force will also in a high-speed engine cause destructive hammering in the eccen-

tric strap, so soon as wear has developed lost motion. A constant pressure on top, however, will absorb the momentum, and keep the wearing parts in unchanging contact. All stress in the moving parts of the gear is by this means converted into compression, which state of things has the incidental advantage of lessening the "fatigue" in the materials used, for the reason that no stress is reversed in direction.

A summary of the various characteristics possessed by this valve gear brings out a very flattering result:

Minimum number of parts and simplicity of form and construction; a free inlet and release, and a cut-off so sharp as to rival even a Corliss; a balanced valve of the piston type, with very small clearance spaces, and which by virtue of its position is easily kept steam tight; great facility for varying the cut-off; perfect adaptability to the difficulties of high rotative speed.

Such are in brief its leading advantages. I can think of but two shortcomings. One is that the limit of its general adoption in the form here represented is reached in an engine having single-acting cylinders only. The other is the difficulty of arranging it to run reversing. The first objection, and perhaps the second as well, will have but slight effect on its destiny, insomuch as its proper field of action will be confined to the high-speed engine where single-acting cylinders will in all probability prevail.

DISCUSSION.

Mr. A. F. Nagle.—It occurs to me to suggest that this paper would receive great additional value if the author would accompany it with an analysis of the valve gear, showing the opening of the ports in proportion to the piston movement, in particular. It has been claimed for this engine by some that the valve action is quicker than any ordinary engine, and if this is so, it should have been demonstrated by the necessary diagram. Some investigations of my own have induced me to the belief that the valve opening is the same as that of the ordinary slide valve operated by the usual eccentric motion.

I would suggest, further, that the indicator card should be supplemented by data concerning the size of cylinder, piston stroke, port areas, clearances, etc. I would like to see, also, the combined diagram with the theoretical curve drawn thereon.

Mr. Walworth.—I would like to ask the writer what velocities are used in these engines.

Mr. Svenson.—In regard to Mr. Nagle's remark I would simply say that I rather hesitated to go into a lengthy description of the valve gear. Complicated diagrams I thought unnecessary. I should have been very pleased to present them, had I thought they would be asked for.

In regard to the speeds, I would say that the rotative speed is 500 revolutions per minute in a small engine, say of 50 horse-power, and 300 revolutions in an engine of 500 horse-power. The stroke is very short, being only six inches in a 100 horse-power engine. The piston speed is low. The average piston speed is 500 feet per minute. Of course, the low speed brings about those various advantages, such as reduced wear, dangers from water, and so forth, that the other high-speed engines having high rotative speed do not possess.

In replying further to the criticisms of Mr. Nagle, the author may be permitted to remark that while it is true that the beginning of the steam inlet openings are quite similar in this gear to that of an ordinary slide valve, yet it should be carefully remembered that the valve has nothing to do with the closing of the inlet ports, so that it, therefore, needs not, and it does not, begin its return stroke immediately after full opening of the inlet ports, as is the case in an ordinary slide valve. Hence it follows that the inlet area is not gradually decreased towards cut off, as in an ordinary slide valve gear. It is this peculiarity which makes the end of the admission period so different from that produced by the valve motion referred to by Mr. Nagle.

I will also take the opportunity to note that the travel of the valve is constant, no matter at what point the steam is cut off. This means not only full port opening at the early cut-offs, but also the absence of that uneven wear of the valve face which soon makes an ordinary slide valve leaky.

DCCXCIV.*

THE GENERATION AND UTILIZATION OF STEAM BY THE LYKENS VALLEY COAL COMPANY AND SUMMIT BRANCH COAL COMPANY, DAUPHIN COUNTY, PA.

BY R. V. NORRIS, WILKESBARRE, PA.
(Member of the Society.)

INTRODUCTION.

THE following report was made at the instigation of Mr. Morris Williams, manager Pennsylvania Railroad Coal companies, and is published with his permission. The original intention was to discover and, if possible, to remedy a large discrepancy in the amount of coal used for firing at the two collieries, and to point out the directions in which improvements could be made.

No attempt was made to attain scientific accuracy, but the work was done as well as practicable with the appliances at hand. The plants examined represent rather better than average anthracite practice, especially as to the boilers, a considerable proportion of which are of the water-tube type. The fuel was "Lykens Valley red ash" buckwheat coal, a semi-anthracite, and with excellent steaming properties, which accounts for the high evaporation of the cylinder boilers.

The practical results of the work thus far have been the shutting off of twenty cylinder boilers at the Williamstown and six at the Bear Valley Shaft plant, the 833 horse-power of the former representing the saving of steam due to repairing the lines, setting valves, etc., no other changes having been made.

Maps and Tables.—The following reproductions of maps and tables are attached:

Map A. L. V. C. Co., map of outside steam lines.
" B. L. V. C. Co., map of inside steam lines.
" C. S. Br. C. Co., map of outside steam lines, Williamstown.

* Presented at the New York meeting (November, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

Map D. S. Br. C. Co., map of inside steam line, Williamstown.
" E. S. Br. C. Co., map of steam lines, Bear Valley.
Table No. 1. Schedule of dimensions of boilers, L. V. C. Co.
" 2. " " " " " S. Br. C. Co.
" 3. Boiler tests, L. V. C. Co. and S. Br. C. Co.
" 4. Comparative test of blower and fan draught.
" 5. Engines and pumps, sizes and performance, L. V. C. Co.
" 6. " " " " " " S. Br. C. Co.
" 7. Indicator cards, engines and pumps, L. V. C. Co.
" 8. " " " " " S. Br. C. Co.
" 9. Steam lines, lengths, and condensation losses, L. V. C. Co.
" 10. " " " " " " S. Br. C. Co.
" 11. Steam generated and used, L. V. C. Co. and S. Br. C. Co.
" 12. Table of friction and condensation losses in steam lines.

Boilers.—The boiler plants examined and tested comprised in all 56 cylinder boilers and 540 horse-power of Babcock & Wilcox boilers, belonging to the Lykens Valley Coal Company, and 66 cylinder boilers, 300 horse-power of return tubular and 2,300 horse-power of Babcock & Wilcox boilers, belonging to the Summit Branch Coal Company. The details of sizes and location of these are given in Tables Nos. 1 and 2 attached.

Boiler Tests.—Extended tests, each of several days' duration, under ordinary working conditions, were made on all the important plants (see Table No. 3), and indicator cards were taken from all steam cylinders supplied by each plant during its test.

Fuel and Ashes.—The fuel and ashes were measured in cars struck off level full, and the weight per cubic foot determined by samples of ten to twenty-seven cubic feet, dried and packed under the same conditions under which the measurements were made, which accounts for the large variation in weight per cubic foot, varying from 47.4 pounds in the first Williamstown test, where the cars were levelled off at the loading point, to 55.4 pounds, when they were transported to the boiler houses before levelling off.

Water.—The total water for tests of the large Lykens Valley and Williamstown plants was measured in tanks and divided in the Lykens Valley test by meters on the Babcock & Wilcox boilers, and in the Williamstown tests by meters on the two Climax boilers and counters on the high and low pressure feed pumps; the meters and counters were read at starting and quitting time each day, and the slip of the high and low pressure pumps determined by solving simultaneous equations obtained from these intermediate readings, and the total water measured

TABLE 1.

Dimensions of Boilers—Lykens Valley Coal Company.

Colliery.	Boiler House.	Style of Boiler.	Number of Boilers.		Dimensions of Boilers.					Heating Surface.		Dimensions of Grates.				H.-P. 34½ Lbs. Water from and at 212° per Hour.		Remarks.
												Per Nest.			Per Group.			
			In Group.	In Nest.	Diameter.	Length.	Number Tubes.	Diam. Tubes.	Length Tubes.	Per Boiler.	Per Group.	Width.	Depth.	Square Feet.	Square Feet.	Per Boiler.	Per Group.	
Lykens..	Bull Engine	Cylinder..	...	...	....	.....	...	...	.....	.....	.....	.....	.....	....	.....	...	...	Taken out May, 1898.
" ..	" "	"	12	2	34"	30' 2"	...	...	....	123.7	1,484	5 10"	6' 0"	35.0	210.0	...	...	
" ..	" "	"	12	3	30	42 0	...	....	.....	156.	1,872	8 9	6 0	52.5	220.0	...	...	
" ..	" "	B. & W...	2	1	36	18 0	63	4"	17' 8"	1,379.	2,758	4 5	6 0	26.5	53.0	120	240	Builder's rating.
" ..	" "	" ...	2	1	36	21 0	34	4	16	1,752.	3,504	7 4	6 0	44.0	88.0	150	300	" "
" ..	New B. H...	Cylinder..	6	3	34	42 2	.	....	.....	176.7	1,060	9 8	7 8	74.1	148.2	...	...	
" ..	" "	"	6	3	34	42 5	...	....	.....	177.8	1,067	9 8	7 8	74.1	148.2	...	...	
" ..	" "	"	6	3	34	42 0	...	....	.....	176.	1,056	9 8	7 8	74.1	148.2	...	...	
" ..	" "	"	6	3	36	42 0	...	....	.....	189 2	1,135	10 3	7 8	78.6	157.2	...	...	
	Wiconisco Pump House	Cylinder.	6	3	30	42 0	...	....	.....	156.	936	8 9	7 8	67.1	134.2	...	...	

TABLE 2.

Dimensions of Boilers—Summit Branch Railroad Company.

Colliery.	Boiler House.	Style of Boiler.	Number of Boilers.		Dimensions of Boilers.					Heating Surface.		Dimensions of Grates.				Horse-power 34½ lbs. Water from and at 212° per Hour.		Remarks.
												Per Nest.			Per Group.			
			In Group.	In Nest.	Diam.	Length.	Number Tubes.	Diam. Tubes.	Length Tubes.	Per Boiler.	Per Group.	Width.	Depth.	Sq. Ft.	Sq. Ft.	Per Boiler.	Per Group.	
Williamstown.	Williamstown.	Cylinder	12	3	34″	42′ 0″	.....	....	.....	175.9	2,111	9′ 8″	6′ 0″	58.	232.	.....	.... ..	
,,	,,	,,	24	3	30	42 0	.....	....	.....	156.0	3,744	8 9	6 0	52.5	420.	.....	.....	
,,	,,	,,	18	3	34	42 0	.....	...	.....	175.9	3,167	9 8	6 0	58.	348.	.....	.....	
,,	,,	,,	6	3	34	41 6	.....	....	.....	173.2	1,042	9 8	6 0	58.	116.	.....	.....	
,,	,,	Climax..	2	1	42	22 8	400	3″	10′ 6″	3,119.	6,238	.....	.. ..	.....	... ...	400	800	) Builder's rating. ; Using waste heat.
,,	,,	B. & W.	2	1	36	18 0	63	4	17 8	1,378.	2,758	4 5	6 0	26.5	53.	120	240	Builder's rating.
,,	,,	,,	4	1	42	23 0	112	4	18 0	2,880.	11,520	9 8	7 0	67.7	270.8	250	1,000	,, ,,
,,	Bear Valley : Shaft.....	Cylinder	6	3	34	42 0	.....	....	... ...	175.9	1,056	9 8	7 8	74.11	148.22	.....	.....	
	,,	B. & W.	2	1	36	18 0	63	4	17 8	1,378.	2,758	4 5	6 0	26.5	53.	120	240	Builder's rating.
	,,	,,	2	1	36	21 0	84	4	15 7	1,750.	3,504	7 4	6 0	44.	88.	150	300	,, ,,
	Slope.....	Return Tubular.	2	1	72	18 0	72	3	18 0	1,380.	2,760	6 0	6 0	36.	72.	150	300	,, ,,
	,,	Cylinder	6	3	30	42 0	.....	..	.....	156.	936	8 9	6 0	52.5	105.	.....	.....	Not in use.
,,	Summit Slope.	,,	2	2	30	30 6	.....	....	.....	110.8	222	5 1	6 0	30.5	30.5	.....	.....	,, ,,
,,	Greenfield Pump	B. & W.	1	1	36	18 0	63	4	17 8	1,378.	1,378	4 5	6 0	26.5	26.5	120	120	Builder's rating.
,,	,,	,,	1	1	36	21 0	84	4	15 7	1,750.	1,750	7 4	7 0	51.3	51.3	150	150	,, ,,
Big Lick.....	Big Lick.....	,,	1	1	42	23 3	112	4	18 0	2,880.	2,880	9 8	7 0	67.7	67.7	250	250	,, ,,

in the tanks on which intermediate readings were taken at the same time.

The water for the smaller tests of Bear Valley shaft and Slope and of the Big Lick plants was measured by meters.

All three meters, 1½-inch by 3-inch Trident and 1½-inch Worthington, were tested during the trials and the corrected results used; the tanks were also tested for leakage and corrections made for same, which proved to be very slight.

The methods adopted of dividing the feed-water are, of course, open to criticism, but they seemed the best with the appliances available; the 3-inch Trident meter, being borrowed after the Williamstown tests, was not available for them.

Calorimeter.—It was impracticable to make calorimeter determinations of the quality of the steam at regular intervals; a few were made, however, and averaged. The Bear Valley shaft cylinder boilers are known to have primed badly at times during the test, and the evaporation found is probably high; this priming was also evinced by the very wet steam, $19\frac{3}{10}$ per cent. moisture at the foot of Summit Slope.

Evaporation.—The various plants tested gave evaporations per pound of combustible from and at 212 degrees, as follows:

	B. & W.	Cylinder.	Ret. Tub.
Williamstown (breaker coal)	9.22	7.53	
Williamstown (separator coal)	10.54	6.69	
Bear Valley Shaft	10.00	8.13	
Bear Valley Slope			9.44
Big Lick Slope	7.54		
Lykens	9.22	7.75	

It is noticeable that the Babcock and Wilcox boilers, at Williamstown and Lykens, gave exactly the same evaporation during tests of several days, duration, though the Lykens boilers were worked 18 per cent. over and the Williamstown 18 per cent. under their rating. The very low evaporation of the Big Lick plant is unquestionably due to the water of condensation from 1,250 feet of 8-inch pipe and 1,250 feet of 2-inch pipe running back into the boilers; this, amounting to about 550 pounds per hour, was reevaporated from the temperature of the steam, which would increase the evaporation from 7.54 pounds to about 8.37 pounds, which, considering that the boilers were run but about 44 per cent. of their rating, is not so bad.

The total horse-power (boiler) of the Bear Valley Shaft plant

TABLE 3.—BOILER TESTS, APRIL AND MAY, 1898.

Summit Branch Coal Company.

	Williamstown B. H.—Separator Coal.				Bear Valley Shaft Boiler House—Breaker Coal.									Bear Valley Slope—Breaker Coal.			Greenfield Pump.	Big I Slo
P. B. & W. ax.	Cylinders.	B. & W.	Climax.	Total.	6 Cylinders, 34″ × 42′ 0″.			540 H.-P. Babcock & Wilcox. 1-300 H.-P.; 1-240 H.-P.			Entire Plant, 6 Cylinders and 540 H.-P. B. & W.			300 H.-P. Return Tubular Boilers, 2-150 H.-P. each. V. I. W.			B. & W. 1-150 H.-P. (1-120 H.-P. not in use).	B. & 1-250
Total.	Idle.	Idle.	Idle.	Idle.	Working.	Idle.	Total.	Working.	Idle.	Total.	Working.	Idle.	Total.	Working.	Idle.	Total.	Working.	Work
	May 1, 8.3	0 A.M., to	May 2, '98	, 6.30 A.M.			May 4, 18	98, 11.30 A	.M., to Ma	y 9, 1898.	11.30 A.M.			May 6, 12	M., to Ma	y 9, 12 M.	Furnished by H. McK.	May 19, '98 to 2 p.
122	22	22	22	22	44	76	120	44	76	120	44	76	120	22	50	72	8	
1,440	1,116	324	.....	1,440	.....	.....	148	.....	.....	141	.....	.....	280	.....	.....	72	51.3	67
30,580	10,064	14,278	6,238	30,580	.....	.....	1,056	.....	.....	6,262	.....	.....	7,318	.....	.....	2,760	1,365	2,8
21 5/6 to 1	9 to 1	44 to 1	.....	21 5/6 to 1	.....	.....	7 3/16 to 1	.....	.....	44 5/6 to 1	.....	.....	25 5/6 to 1	.....	.....	38 3/16 to 1	38 1/16 to 1	42 5/6 to
.....	80	120	120	.....	75	75	75	100	100	100	.....	.....	.....	75	75	75	113.2	85
.....	70	122	50	.....				Exhaust steam heating.			.....	.....	.....	.....	.....	.....	Exht. steam.	.....
.....	136	122	122	.....	119	119	119	119	119	119	.....	.....	.....	50	50	50	199.5	
2,287,239	270,213	57,280	.....	327,593	.....	.....	104,996	.....	.....	204,313	.....	.....	309,309	.....	.....	60,934	3,181	2,9
424,475	.....	.....	.....	45,641	Some coal as used Williamstown Boiler House—1st test.						Refuse taken the same.			.....	.....	.....	435	4
18.47	.....	.....	.....	13.93	.....	.....	18 1/4	.....	.....	18	.....	.....	18 1/4	.....	.....	18 1/4	13 7/16	
1,864,824	232,572	49,387	.....	281,959	.....	.....	85,572	.....	.....	166,515	.....	.....	252,087	.....	.....	49,661	2,798	2,5
18,748	12,282	2,608	.....	14,890	.....	.....	875	.....	.....	1,703	.....	.....	2,578	.....	.....	846	398	5
15,286	10,571	2,245	.....	12,816	.....	.....	713	.....	.....	1,388	.....	.....	2,101	.....	.....	690	350	5
.....	.....	.....	.....	164.1	.....	.....	.....	.....	.....	.....	27.7	27.7	.....	9.5	8.2	.....	4 1/4	5 work,
5,952	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	829	.....	.....	277	128	1
.....	3	3	0	.....	10	10	10	3	3	3	.....	.....	.....	3	3	3	1.55	3
14,482,271	1,354,630	471,818	481,244	2,307,692	283,803	439,958	693,761	519,132	991,396	1,510,528	802,935	1,401,354	2,204,289	149,389	252,540	401,929	22,251	16,4
13,901,017	1,313,991	457,603	481,244	2,225,808	255,423	368,962	624,385	503,558	961,654	1,465,212	758,981	1,330,616	2,089,597	144,907	244,964	389,871	21,906	15,9
113,942	59,737	20,803	21,875	102,405	5,805	4,855	5,203	11,445	12,653	12,210	17,250	17,508	17,413	6,587	4,899	5,415	2,738	3,1
.....	1,433,448	409,272	525,000	2,457,720	126,970	116,520	.....	290,384	305,672	.....	417,354	422,192	.....	135,144	117,576	.....	65,712	51,8
81,069,264	.....	.....	.....	.....	.....	.....	3,746,400	.....	.....	8,803,248	.....	.....	12,549,648	.....	.....	3,972,312	.....	.....
16,502,030	1,552,086	520,500	*616,526	2,659,114	288,143	416,226	704,369	572,445	1,093,208	1,665,653	816,588	1,569,494	2,370,022	174,309	294,667	468,976	23,135	19,1
135,262	70,549	23,659	28,024	120,869	6,548	5,477	5,870	13,010	14,384	13,880	19,558	19,861	19,750	7,923	5,893	6,514	2,890	3,8
6.08	4.86	7.98	.....	6.79	.....	.....	5.95	.....	.....	7.17	.....	.....	6.76	.....	.....	6.40	6.67	5.
7.22	5.75	9.06	.....	8.12	.....	.....	6.71	.....	.....	8.15	.....	.....	7.66	.....	.....	7.70	7.05	6.
7.45	5.65	9.27	.....	8.01	.....	.....	7.29	.....	.....	8.80	.....	.....	8.29	.....	.....	7.85	7.85	6.
8.85	6.69	10.54	.....	9.43	.....	.....	8.13	.....	.....	10.09	.....	.....	9.40	.....	.....	9.44	8.26	7.
6.42	4.85	7.64	.....	6.84	.....	.....	5.89	.....	.....	7.25	.....	.....	6.82	.....	.....	6.84	5.99	5.
13.0	11.0	8.0	.....	10.3	.....	.....	6.0	.....	.....	12.1	.....	.....	8.9	.....	.....	11.8	7.8	8.
4.3	7.0	1.7	4.5	3.9	6.2	5.2	5.6	2.1	2.3	2.2	2.7	2.7	2.7	2.9	2.1	2.4	2.1	1.
91.8	63.2	73.0	.....	84.0	44.2	37.0	39.7	92.3	102.0	98.4	67.7	68.7	68.4	110.0	81.8	90.5	56.3	56.
3,949	2,049	687	*813	3,549	189	156	170	378	418	402	567	574	572	230	171	189	84	1
.....	.....	1,240	800	.....	.....	.....	.....	540	540	540	.....	.....	.....	300	300	300	150	2
.....	5 7/16	.....	11.2	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
7.74	4.91	20.8	7.67	8.62	.....	.....	6.21	.....	.....	15.6	.....	.....	12.6	.....	.....	14.6	16.3	26
0.36	0.54	0.47	.....	0.41	.....	.....	0.81	.....	.....	0.35	.....	.....	0.51	.....	.....	0.38	0.61	0.

† Combination 24 cylinders and 2 Climax boilers. These 24 cylinders burned approximately 40 per cent. more coal than balance of cylinders economic evaporation calculated from this assuming const

Big Lick Slope.	Lykens Valley Coal Company.										Remarks.
	L. V. C. Co. "Bull" and "New" Boiler Houses—Breaker Coal.									L. V. C. Co. Fresh Water Pumps.	
B. & W. 250 H.-P.	48 Cylinders. 18: 34″ × 42′ 2″. 12: 31″ × 30′ 2″. 12: 30″ × 42′ 0″. 6: 36″ × 42′ 0″.			540 H.-P. Babcock & Wilcox. 1-300 H.-P.; 1-240 H.-P.			Entire Plant. 48 Cylinders and 540 H.-P. B. & W.			6 Cylinders. 30″ × 42′ 0″.	
Working.	Working.	Idle.	Total.	Working.	Idle.	Total.	Working.	Idle.	Total.	Not tested. Approx.d estimate from coal use	
r 19, '98, 9 A.M. to 2 P.M.			May 10,	1898, 6.30 A.M., to M	ay 16, 1898	, 6.30 A.M.					The Williamstown plant, total water measured in tanks, divided by meters on Climax and counters on pumps; slip of high-pressure pump 8.70 per cent., and of low-pressure pump 11.66 per cent., determined by algebraic calculation. Bear Valley Shaft and Slope, and Big Lick plants, water measured by tested meters. Lykens Valley plant, total water measured in tanks, divided by meters on B. & W. Feed in "Bull" boiler house heated by exhaust steam, in "New" boiler house by pipe heaters in flues. Average feed temperatures for cylinder boilers calculated. Coal and refuse measured in cars, and weight per cubic foot ascertained for dry coal and ashes under similar conditions.
5	55	89	144	55	89	144	55	89	144		
67.7	.....	.....	1,032	.....	.....	141	.....	.....	1,173	134	
2,880	.....	.....	7,674	.....	.....	6,362	.....	.....	13,936	936	
62½ to 1	.....	.....	7.4 to 1	.....	.....	44.4 to 1	.....	.....	11.9 to 1	7 to 1	
85.7	80	80	80	115	115	115	.....	.....	.....		
.....	104	104	104	Heated	by exhaust steam.		.....	.....	.....		
55	148	148	148	171	171	171	.....	.....	.....		
2,973	.....	.....	1,597,528	.....	.....	423,201	.....	.....	2,020,729		
430	.....	.....	315,027	.....	.....	80,323	.....	.....	395,350		
14½	.....	.....	19.71	.....	.....	18.98	.....	.....	19.56		
2,543	.....	.....	1,282,501	.....	.....	342,878	.....	.....	1,625,379		
595	.....	.....	11,094	.....	.....	2,939	.....	.....	14,033		
509	.....	.....	8,906	.....	.....	2,381	.....	.....	11,287		
work, 2 idle.	.....	.....	.....	.....	.....	.....	154.3	130.5	.....	Wint'r 13½, Sum'r 9¼	
120	.....	.....	.....	.....	.....	.....	.....	.....	4,486	345	
3 1	3	3	3	3	3	3	.....	.....	.....		
16,487	3,930,002	4,989,846	8,919,848	1,467,598	1,538,130	3,005,728	5,397,600	6,527,976	11,925,576		
15,976	3,312,102	4,840,151	8,652,253	1,423,570	1,491,986	2,915,556	5,235,672	6,332,137	11,567,809		
3,195	63,311	54,384	60,085	25,882	16,764	20,247	95,193	71,148	80,332	5,730	
51,800	1,468,413	1,305,216	.....	502,634	402,336	.....	1,971,047	1,707,552	.....	137,500	
.....	.....	.....	43,073,208	.....	.....	14,477,232	.....	.....	57,550,440		
19,171	4,376,293	5,556,493	9,932,786	1,545,285	1,619,551	3,164,836	5,921,578	7,176,044	13,097,622		
3,834	79,569	62,432	68,977	28,096	18,197	21,978	107,665	80,629	90,955		
5.37	.....	.....	5.51	.....	.....	6.89	.....	.....	5.72	5.5	
6.45	.....	.....	6.33	.....	.....	7.48	.....	.....	6.48		
6.28	.....	.....	6.75	.....	.....	8.50	.....	.....	7.12		
7.54	.....	.....	7.75	.....	.....	9.22	.....	.....	8.06		
5.47	.....	.....	5.62	.....	.....	6.69	.....	.....	5.84		
8.8	.....	.....	10.8	.....	.....	20.8	.....	.....	12.0		
1.3	10.4	8.1	9.1	4.5	2.9	3.5	7.7	5.9	6.5		
56.7	77.1	60.5	66.8	199.3	128.4	155.9	91.8	68.8	77.6		
111	2,308	1,811	2,001	815	528	638	3,123	2,339	2,639	200	
250	.....	.....	.....	540	540	540	.....	.....	.....		
.....	.....	.....	3⅝	.....	.....	.....	.....	.....	.....		
26.0	.....	.....	3.84	.....	.....	9.81	.....	.....	5.38		
0.61	.....	.....	0.52	.....	.....	0.22	.....	.....	0.44		

Plant.	Were Meas'd.	W't Coal, cu. ft.	W't Ash, cu. ft.
W'mst'n 1st..	Loading pt.....	47.4	40.5
" 2d..	Boiler house ...	51.2	46.0
Bear Valley..	Boiler houses..	54.6	.....
Lykens	Boiler houses..	55.4	51.8

shak'n down.

All "Breaker" coal averages: 47¼ per cent. No. 2 Buck, 52½ per cent. No. 3 Buck. "Separator" coal averages about 5 per cent. Nut, 20 per cent. Pea, 35 per cent. No. 1 Buck, 40 per cent. No. 2 Buck.

constant rate for cylinders.

Item.	Working or Idle Time at Colliery.		60 Cylinders, 30: 34″ × 42″ 0″. 6: 34″ × 41 24: 30″ × 42″ 0″. Working.	Idle.	Tot
1	Date of test				
2	Duration of test	Hours.	53½	68½	12
3	Grate surface	Sq. ft.			
4	Heating surface	Sq. ft.			1
6	Ratio heating to grate surface				9 t
7	Steam pressure by gauge	Lbs.	80	80	8
15	Feed-water temperature before heater	Deg. F.	66	66	6
15a	Feed-water temperature after heater	Deg. F.	122	122	12
18	Dry coal consumed	Lbs.			1,71
19	Total refuse dry	Lbs.			
19a	Total refuse	P. c.			
20	Total combustible dry	Lbs.			1,39
21	Dry coal consumed per hour	Lbs.			1
22	Combustible consumed per hour	Lbs.			1
	Dry coal consumed per day	Tons.			
	Dry coal consumed per month (24 w., 6 idle)	Tons.			
24	Moisture in steam, by calorimeter	P. c.	5	5	5
26	Total weight of water pumped into boilers	Lbs.	4,589,956	4,738,060	9,32
27	Water actually evaporated, corrected for quality of steam	Lbs.	4,360,458	4,501,157	8,86
	Water actually evaporated per hour	Lbs.	81,504	65,711	7
	Water actually evaporated per day (working : 11 w., 13 idle)	Lbs.	1,750,787	1,577,064	
	Water actually evaporated per month (24 w., 6 i. days)	Lbs.			51,51
28	Equivalent water evaporated into dry steam, from and at 212 degrees	Lbs.	5,177,608	5,344,674	10,52
30	Equivalent water evaporated, from and at 212 degrees per hour	Lbs.	96,778	78,025	8
31	Water evaporated per pound dry coal, actual temperatures and pressure	Lbs.			5
32	Water evaporated per pound dry coal, from and at 212 degrees	Lbs.			6
	Water evaporated per pound combustible, actual temps. and pressure	Lbs.			6
33	Water evaporated per pound combustible, from and at 212 degrees	Lbs.			7
34	Commercial evaporation : equivalent water evaporated per pound dry coal less refuse, 70 lbs. gauge from 100 deg. feed-water	Lbs.			5
35	Dry coal burned per square foot grate surface per hour	Lbs.			12
39	Water evaporated, from and at 212 deg. per sq. ft. heating surface per hour	Lbs.	9.6	7.8	8
40	Water evaporated, from and at 212 deg. per sq. ft. grate surface per hour	Lbs.	86.7	70.0	77
43 41	Horse-power—basis : 30 lbs. water per hour, 70 lbs. steam from 100 deg. feed, 34½ lbs. water, from and at 212 degrees		2,806	2,262	
	Horse-power : builder's rating				
	Per cent. horse-power due to feed-water heater	P. c.			4
	Square feet heating surface per horse-power	Sq. ft.			4
	Grate surface per horse-power	Sq. ft.			0

WILLIAMSTOWN BOILER HOUSE—BREAKER COAL.										WILLIAMSTOWN B. H.—SEPAR		
″×41′ 6″.	1,240 H.-P. Babcock & Wilcox. 2-500 H.-P.; 1-240 H.-P.		Waste Heat Climax. 2-400 H.-P. = 800 H.-P.			Entire Plant. 60 Cylinders: 1,240 H.-P. B. & W. 800 H.-P. Climax.			Cylinders.	B. & W.	Climax	
Total.	Working.	Idle.	Total.	Working.	Idle.	Total.	Working.	Idle.	Total.	Idle.	Idle.	Idle.
		April 26, 1898,	6.30 A.M.,	to May 1,	1898, 8.30	A.M.				May 1, 8.30 A.M., to		May 2, ′9
122	53½	68½	122	53½	68½	122	53½	68½	122	22	22	22
1,116	.....	.....	324	.....	.....	.....	.....	.....	1,440	1,116	324	.....
10,064	.....	.....	14,278	.....	.....	6,238	.....	.....	30,580	10,064	14,278	6,2
9 to 1	.....	.....	44 to 1	.....	.....	.....	.....	.....	21⅞ to 1	9 to 1	44 to 1	.....
80	120	120	120	120	120	120	.....	.....	.....	80	120	120
66	102	102	102	50	50	50	.....	.....	.....	70	122	50
122	102	102	102	102	102	102	.....	.....	.....	136	122	122
1,714,742	.....	.....	572,497		Waste he at.		.....	.....	2,287,239	270,213	57,280	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....	422,475	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....	18.47	.....	.....	.....
1,398,058	.....	.....	406,766	.....	.....	.....	.....	.....	1,864,824	232,572	49,387	.....
14,055	.....	.....	4,693	.....	.....	.....	.....	.....	18.748	12,282	2,608	.....
11,460	.....	.....	3,826	.....	.....	.....	.....	.....	15,286	10,571	2,245	.....
.....	.....	.....	.....	.....	.....	.....	201.5	186	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....	5,952	.....	.....	.....
5	3	3	3	0	0	0				3	3	0
9,328,016	1,696,072	2,132,375	3,828,447	666,810	658,998	1,325,808	6,952,858	7,529,433	14,482,271	1,354,630	471,818	481,2
8,861,615	1,645,190	2,068,404	3,713,594	666,810	658,998	1,325,808	6,672,458	7,228,559	13,901,017	1,313,991	457,603	481,2
72,636	30,751	30,196	30,439	12,464	9,606	,0,867	124,709	105,513	113,942	59,727	20,863	21,8
.....	733,789	724,704	.....	261,982	240,544	.....	2,743,558	2,532,312	.....	1,433,448	499,272	525,0
51,511,272	.....	.....	21,887,160	.....	.....	7,670,832	.....	.....	81,069,264	.....	.....	.....
10,522,282	1,965,624	2,395,872	4,301,496	*893,519	*784,733	*1,678,252	7,976,751	8,525,279	16,502,030	1,552,086	520,500	*616,5
86.218	35,619	34,976	35.258	16,701	11,456	13,756	149,098	124,456	135,262	70,549	23,659	28,0
5.16	.....	.....	6.48	.....	.....	.....	.....	.....	6.08	4.86	7.98	.....
6.13	.....	.....	7.51	.....	.....	48.02	.....	.....	7.22	5.75	9.06	.....
6.34	.....	.....	7.96	.....	.....	.....	.....	.....	7.45	5.65	9.27	.....
7.53	.....	.....	9.22	.....	.....	49.83	.....	.....	8.85	6.69	10.54	.....
5.45	.....	.....	6.68	.....	.....	.....	.....	.....	6.42	4.85	7.64	.....
12.6	.....	.....	14.5	.....	.....	.....	.....	.....	13.0	11.0	8.0	.....
8.6	2.5	2.4	2.44	2.6	1.8	2.2	4.8	4.1	4.3	7.0	1.7	4.5
77.3	109.9	107.9	108.7	.....	.....	.....	103.5	86.4	91.8	63.2	73.0	.....
2,500	1,031	1,013	1,021	*484	*384	428	4,321	3,659	3,949	2,049	687	*813
.....	1,240	1,240	1,240	800	800	800	.....	.....	.....	.....	1,240	800
.....	.....	.....	.....	.....	.....	12.6	.....	.....	.....	5⅞	.....	11.2
4⅛	.....	.....	.....	.....	.....	14.6	.....	.....	7.74	4.91	20.8	7.6
4.03	.....	.....	14.0	.....	.....	.....	.....	.....	0.36	0.54	0.47	.....
0.45	.....	.....	0.32	.....	.....	.....	.....	.....				

* Including heating feed-water for B. & W. boilers.

† Combination 24 cylinders and 2 Cli

was but 572, which could be readily taken care of by the 540 horse-power of Babcock and Wilcox boilers, allowing the less economical cylinder boilers to remain idle. (This has since been done.)

The evaporation of the combination of 24 cylinder boilers, with two 400 horse-power Climax boilers utilizing the waste heat, is only a rough approximation, no attempt having been made to test these 24 cylinders apart from the other 36. They consumed, approximately, 40 per cent. more coal than any other 24 boilers, and in getting at the economy of the combination it was assumed that the cylinder boilers maintained the average evaporation per pound of coal of the whole 60 tested.

Blowers.—It will be noted in Table No. 11 that a very large amount of steam is consumed by the blowers. The quantity, 830 pounds steam per ton coal, is based on tests made by Messrs. McKay and Bauer, Table No. 4. This could be reduced to at least one-third by the use of fan draught, and in most, if not all cases, there appears to be sufficient exhaust steam available within reasonable distance to turn into the blast and prevent burning out of the grate-bars. The experiments referred to were made with a fan of very low efficiency, and even that showed a great difference, over 50 per cent., in its favor.

Safety Valves.—The leakage of safety valves is of sufficient importance to make an item in Table No. 11. It seems impracticable to prevent some leakage from safety valves of the weighted type; the remedy would be either to use spring valves or to utilize the escaping steam either to heat feed-water or to turn into a fan blast.

Feed-heating.—The feed-water for one of the Lykens Valley and the Bear Valley shaft boiler houses is heated by exhaust steam, much hotter than any other feed; at the Williamstown boiler house the exhaust from the conveyer engine and low-pressure feed-pump is utilized to assist the heating from stack tanks, and such heating has been allowed for in the boiler tests. Wherever any exhaust or escaping steam is available it should be utilized for feed-heating, and even though a very considerable length of exhaust pipe may be required it will be found profitable.

TABLE 11.

Steam Generated and Used—Summit Branch Coal Company and Lykens Valley Coal Company Boiler Plants.

July 28, 1898.

	Summit Branch Coal Co.										Lykens Valley Coal Co.		
	Williamstown B. H. Breaker Coal.		Williamstown B. H. Separator Coal.		Bear Valley Shaft Boiler House.		Bear Valley Slope Boiler House.		Greenfield B. H.	Big Lick Slope B. H.	Lykens Valley Coal Co. Colliery Plant.		L. V. C. Co. Freshwater Plant.
	Per Working Day 24 Hrs. 11 Work, 13 Idle.	During Boiler Test 122 Hrs. 53¾ Work, 68¼ Idle.	Per Idle Day 24 Hrs.	During Boiler Test 22 Hrs. Idle.	Per Working Day 24 Hrs 11 Work, 13 Idle.	During Boiler Test 120 Hrs. 44 Work, 76 Idle.	Per Work Day 24 Hrs. 11 Work, 13 Idle.	During Boiler Test 72 Hrs. 22 Work, 50 Idle.	Not Tested. Per Day 24 Hrs.	During Boiler Test 5 Hrs. Work.	Per Working Day 24 Hrs. 11 Work, 13 Idle.	During Boiler Test 144 Hrs. 55 Work, 89 Idle.	Not Tested. Per 24 Hrs.
Total dry steam generated.	2,743,600	13,901,000	2,457,700	2,225,900	417,400	2,089,600	136,100	389,900	.....	15,976	1,971,000	11,567,800	137,500
Consumption of Steam:													
Fan engines	201,300	1,023,900	201,300	184,600	37,800	189,000	14,000	42,000	.....	.....	183,100	1,098,600	.....
Hoisting engines	171,500	860,000	33,500	31,000	5,200	26,000	8,500	21,000	.....	13,100	273,200	1,386,100	.....
Misc. outside engines	76,900	386,300	21,100	19,300	.....	.....	.....	.....	.....	.....	157,300	786,500	85,000
Outside pumps	33,300	169,000	30,000	27,500	6,100	30,500	.....	.....	82,000	.....	132,700	689,000	42,400
Mine pumps	867,500	4,410,600	867,500	794,400	205,200	1,031,000	102,000	306,300	.....	.....	727,700	4,363,900	.....
Blowers @ 830 lbs. steam per ton	166,800	847,500	135,900	121,300	22,900	114,600	Natural draft.		.....	Nat. draft.	127,800	748,700	9,500
Cond'sat'n and heat'g bldgs	179,500	912,500	179,500	164,500	24,600	108,300	7,300	19,600	.....	200	81,700	490,200	.....
Leak'ge of saf'ty valves, est.	28,800	146,400	28,800	24,400	6,000	30,000	New boilers.		.....	200	25,000	150,000	.....
Total Steam Accounted for	1,725,600	8,756,200	1,497,600	1,367,000	307,800	1,539,400	131,900	388,900	.....	13,500	1,708,500	9,713,000	136,900
Difference	1,018,000	4,144,800	960,100	858,900	109,600	550,197	4,200	971	.....	2,476	262,500	1,854,800	.....
Per Cent. of Steam Accounted for	62 9/10	63	60 9/10	61	73 7/10	73 3/10	96 5/10	99 7/10	.....	84 5/10	86 8/10	84	.....
Coal used (tons)	201.5	1,021.1	164.1	146.2	27.7	138.1	9.5	27.2	4¼	1.3	154.3	902.1	11.5
Coal not accounted for	74.8	377.8	64.2	57.0	7.3	37.0	0.33	0.08	.....	0.20	20.7	144.3	.....

TABLE 4.

Comparative Test of Three Cylinder Boilers at Short Mountain Colliery with Steam and Fan Blowers.

Fan Blower consisted of Gangway Fan 33″ diameter, four paddles 9″ × 9½″, driven by small slide-valve engine with cylinder 4 1/16″ diameter, 7¾″ stroke. Steam supplied by small upright boiler on which an evaporative test was run during the test on the cylinder boilers. Duration of test, 8 hours.

Steam Blower made of ¾″ pipe, circle 6¾″ diameter, 16 holes, tapered ⅛″ outside, 1/16″ inside diameter. Steam supplied by upright boiler on which test was run as above. Duration of test, 8 hours.

Dimensions of boilers 36″ diameter, 42′ long. Area grate surface, 3 boilers, 61.5 square feet.	With Steam Blower.	With Fan Blower.	Remarks.
Coal.			
Total coal burned (less moisture)	7,700 lbs.	6,100 lbs.	In test with fan blower the exhaust from fan engine was turned into air current, and found sufficient to keep the grates free from clinker.
" ash	1,330 "	7,027 "	
" combustible	6,370 "	5,073 "	
Per cent. of ash	17.2 per cent.	16.8 per cent.	
Coal burned per hour	962.5 lbs.	762.5 lbs.	
Combustible burned per hour	796.3 "	634.1 "	
Water.			
Total water evaporated—actual conditions	39,241 lbs.	34,890 lbs.	
" " " per hour—actual conditions	4,905 "	4,361 "	
Equivalent water evaporated per hour from and at 212°	5,444 "	4,867 "	
Water evaporated per hour per lb. of coal—actual conditions	5.10 "	5.59 "	
" " " " " " " combustible—actual conditions	6.16 "	6.87 "	
" " " " " " " coal from and at 212°	5.66 "	6.38 "	
" " " " " " " combustible from and at 212°	6.84 "	7.67 "	
H.-P. developed per hour from and at 212°	157.81 "	141.1 "	
Average boiler pressure	77 "	78 "	
Average temperature of feed-water	134°	137°	
Blowers.			
Boiler H.-P. used by blowers per hour from and at 212°	11.9 H.-P.	5.64 H.-P.	Average steam pressure at steam blowers = 74 lb.
Per cent. of the developed H.-P. of the three boilers used for blowers	7.4 per cent.	4 per cent.	" I. H. P. of fan engine, 1.62 H.-P.
Cubic feet of air per minute	2,502 ft.	3,506 ft.	" number revolutions of fan engine, 160 rev.
Average water gauge	.44″	.52″	" " " " " " 915 "
Horse-power in air	.173 H.-P.	.28 H.-P.	Per cent.—Useful effect of fan, 17 per cent.
Steam used in blowers per ton fuel	830 lbs.		

Tests made by Mr. F. J. Bauer, Boiler Inspector, and Mr. Hood McKay, Assistant Engineer.

Lykens, Pa., *June* 21-22, 1896.

Copied from *Colliery Engineer*, August, 1897.

Boiler-house Piping.—The piping in the boiler houses was found in excellent condition, except for a few leaky joints which could not be repaired without shutting down the entire plant, and during the Lykens test the whole boiler house was shut down for over two hours on account of a blown-out joint in the main drum. It seems that in the future it would be worth while to try to pipe all boiler houses in duplicate, so that any joint might be gotten at for repairs without shutting down the entire plant; the secondary piping could, if desirable to avoid condensation, be just of sufficient capacity to carry the steam with the maximum allowable loss in pressure.

Power and Coal Consumption.—The total boiler horse-power of the two companies and coal consumed was found to be, under ordinary working conditions:

	Boiler Horse-Power.	Coal per Month, 24 Working Days, 6 Idle Days.
		Tons.
L. V. C. Co.		
Colliery plant	3,123	4,486
Fresh-water plant	200	345
Total	3,323	4,831
S. Br. C. Co.		
Williamstown plant	4,321	5,952
Bear Valley Shaft	567	829
Bear Valley Slope	230	277
Greenfield	84	128
Big Lick	111	120
Total	5,313	7,306

Cost of Steam: per Boiler Horse-power.—From the boiler test the cost of steam per boiler horse-power at the various plants approximates as follows, estimating the buckwheat coal used at 75 cents per ton, including labor of firing and removal of ashes, and the interest, depreciation, and repairs to boiler plants at 15 per cent. on $10 per horse-power.

TABLE 5.—LYKENS VALLEY COAL COMPANY—ENGINES AND PUMPS, MAY

Steam Cylinders.					Water Data.				Hoisting Data.					Speed.		Power and Steam Economy.			
Steam Line.	Separator.	Trap.	Cocks Open=O.; or Closed=C.	Exhaust Into.	Diameter Plunger.	Diam. Column.	Length Column Pipe.	Lift, Vertical including Suction.	Revolutions per Hoist.	Hoists per Day, Test.	Hoists per Day, Maximum.	Hoists per Idle Day.	Steam per Hoist.	Revolutions per Minute, Test.	Revolutions per Minute, Max.	M. E. P., Test.	Horse Power, Test.	Indicated Steam per H.-P. per Hour, Test.	Total Steam per H.-P. per Hour, Incl. Cyl. Con-
					In.	In	Ft.	Ft.	Revs.	No.	No.	No.	Lbs.	Revs.	Revs.	Lbs.	H.-P.	Lbs.	Lbs.
B	Yes	Yes	C.	Atmosphere	....	....	....	.....	.....	.....	....	....	.....	58	58	8.7	41.4	74.0	8
B & E	Yes	Yes	C.	"	....	....	....	.....	.....	.....	....	....	.....	60	60	8.15	41.2	78.6	8
C	Yes	Yes	C.	"	....	....	....	.....	.....	.....	....	....	.....	45	45	23.7	13.7	46.3	5
.....	.....	.....	.....	.....	....	....	....	.....	.....	.....	....	....	.....	.....	.....	.....	96.3	.....	7
C	Yes	Yes	¼ turn o.	Atmosphere	....	....	....	.....	150	C. 50 A. 35	100 40	... 12	1,070 760	60 60	100 60	.....	400.0 192.6	E. 47, W. 50 52.7	E. 53, W 5
B	Yes	Yes	C.	"	....	....	....	.....	170	210	250	10	807	64 130	⅓ 130	25.2	340.0	68.6	7
M & N	Yes	Yes	¼ turn o.	Ditch	....	....	....	.....	128	103	150	8	209	110	.....	27.0	275.0	32.2	3
Boil. H.	No	No	O.	Atmosphere	....	....	....	.....	250	17	25	10	123	102	.....	16.4	22.9	54.4	5
.....	.....	.....	.....	.....	....	....	....	.....	.....	.....	....	....	.....	.....	.....	.....	1,038.	.....	5
A	Yes	Yes	C.	Atmosphere	....	....	....	.....	.....	.....	....	....	.....	65	65	26.1	156.3	40.0	4
A	Yes	Yes	C.	"	....	....	....	.....	.....	.....	....	....	.....	84	84	42.5	79.7	33.8	3
B	....	....	O.	"	....	....	....	.....	.....	.....	....	....	.....	88	88	14.0	44.2	68.6	7
B	....	....	.....	"	....	....	....	.....	.....	.....	....	....	.....	120	.....	10.1	23.3	63.5	17
B	....	....	.....	"	....	....	....	.....	.....	.....	....	....	.....	80	.....	27.4	41.6	46.3	5
C	No	No	O.	"	....	....	....	.....	.....	.....	....	....	.....	130	.....	11.1	5.7	66.0	7
B	No	No	O.	"	....	....	....	.....	.....	.....	....	....	.....	160	.....	12.5	3.4	110.3	11
.....	.....	.....	.....	.....	....	....	....	.....	.....	.....	....	....	.....	160	.....	12.5	3.4	110.3	11
.....	.....	.....	.....	.....	....	....	....	.....	.....	.....	....	....	.....	.....	.....	.....	321.	.....	4
Boil. H.	No	No	C.	Heater	4	....	....	.....	.....	.....	....	....	.....	50	70	23.1	5.0	54.2	5
Boil. H.	No	No	C.	"	4	....	....	.....	.....	.....	....	....	.....	50	70	28.5	4.3	57.0	6
Boil. H.	No	No	¼ turn o.	"	6	....	....	.....	.....	.....	....	....	.....	18	25	48.9	7.3	53.4	5
Boil. H.	No	No	.....	Atmosphere	6	....	....	.....	.....	.....	....	....	.....	17.7	25	33.9	4.8	58.3	6
D	..	....	.....	Heater	12	....	....	.....	.....	.....	....	....	.....	7	.....	18.8	57.7	41.8	4
A	No	No	.....	Column	....	4	....	94	.....	.....	....	....	.....	.....	.....	.....	10.7	.....	70
.....	.....	.....	.....	.....	....	....	....	.....	.....	.....	....	....	.....	.....	.....	.....	89.8	.....	12
Boil. H.	No	No	⅛ turn o. ⅛ turn o.	Heater	21¼	20	550	319	.....	.....	....	....	.....	4.17	4½	37.6	68.4	44.6	4
K	Separator & trap on each level			Syphon condenser	9 3/16	10 12	541 121	{ 318	.....	.....	....	....	.....	19.5	25	31.1	80.5	53.3	5
K			"	Tank	11 5/16	8	568	293	.....	.....	....	....	.....	4	14	27.5	19.7*	58.6	6
K			"	Tail pipe	16¼	14	520	317	.....	.....	....	....	.....	6	8	25.1	61.0*	50.8	5
I & J			"	Syphon condenser	16¼	12	489	331	.....	.....	....	....	.....	3½	9	31.5	39.9	50.9	5
I & J			"	Tank	12½	8	485	331	.....	.....	....	....	.....	6	9	30.6	39.2*	48.0	5
F			"	Syphon condenser	16¼	12	420	328	.....	.....	....	....	.....	5¼	9	27.0	62.2	55.0	5
F			"	"	12 5/8	8 12	492	328	.....	.....	....	....	.....	8.5	12	27.2	56.6	58.8	6
M			"	Ditch	12½	8	325	321	.....	.....	....	....	.....	5	6½	30.8	47.4*	51.4	5
I & J			"	"	12 3/8	10	325	321	.....	.....	....	....	.....	6.5	10	30.1	38.6	54.4	5
F			"	Syphon condenser	16	12	342	326	.....	.....	....	....	.....	5¼	9	27.4	63.0	51.4	5
F			"	"	14½	10	358	326	.....	.....	....	....	.....	8.5	12	33.0	69.9	46.0	4
P & Q			"	Tail pipe	10	10	800	673	.....	.....	....	....	.....	9½	17½	26.75 12.30	46.0* 46.9*	{ 34.3	3
P & Q			"	Syphon condenser	10	10	800	660	.....	.....	....	....	.....	9½	17½	26.8 10.1	46.1 38.5	{ 32.8	3
.....	.....	.....	.....	.....	....	....	....	.....	.....	.....	....	....	.....	.....	.....	.....	569.8*	.....	52
.....	....	..	.....	.....	....	....	....	.....	.....	.....	....	....	.....	.....	.....	.....	2,114.9	.....	57

LYKENS VALLEY COAL COMPANY—FRESH-WATER PUMPING STATION.

.....	No	No	.....	Atmosphere	8	....	....	.....	.....	.....	....	....	.....	30	37½	25.2	12.1	60.7	6
.....	No	No	.....	"	8	....	....	.....	.....	.....	....	....	.....	16	20	20.1	12.3	72.0	7
.....	No	No	.....	"	4½	....	....	.....	.....	.....	....	....	.....	25	.....	33.5	1.3	58.3	6
.....	Yes	Yes	.....	"	....	....	....	.....	.....	.....	....	....	.....	246	246	30.1	103.5	26.3	3
.....	Yes	Yes	.....	"	....	....	....	.....	.....	.....	....	....	.....	300	300	.....	80.0	.....	3
.....	....	....	.....	.....	....	....	....	.....	.....	.....	....	....	.....	.....	.....	.....	197.0	.....	3

MAY, 1898.

OMY.	STEAM CONSUMPTION.							
Total Steam per H.-P. per Hour, Incl. Cyl. Condensation.	Total Steam per Hour, Test.	Total Steam per Hour, Maximum.	Total Steam per Working Day, Test.	Total Steam per Idle Day, Test.	Total Steam per Working Day, Maximum.	Total Steam per Idle Day, Maximum.	Cost per Year of Steam Used Above 40 lbs. per I. H.-P. per Hour.	REMARKS.
Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	$	
81.4	3,370	3,370	80,900	80,900	80,900	80,900	997 00	
86.5	3,564	3,564	85,500	85,500	85,500	85,500	1,092 00	
50.9	695	695	16,700	16,700	16,700	16,700	8 00	
79.1	7,629	7,629	183,100	183,100	183,100	183,100	2,097 00	
53, W. 65								
58.0	8,010	13,740	80,100	9,100	137,400	9,100	533 00	Single slope, steam per hoist for round trip.
75.5	16,947	20,175	169,500	8,100	201,800	8,100	1,685 00	Double slope.
35.0	2,152	3,130	21,500	1,700	31,300	1,700	.....	"
59.9	86	130	2,100	1,200	3,100	1,200	.....	Lower by hook lever. Steam for round trip.
58.0	27,195	37,175	273,200	20,100	373,600	20,100	2,218 00	
45.0	7,034	7,034	70,300	.....	70,300	.....	166 00	"Poore" valve.
37.2	2,965	2,965	29,600	.....	29,600	.....	.....	
75.5	3,337	3,908	33,400	.....	39,100	.....	334 00	Max. work, 60.4 H.-P., with 58.8 lbs. ind. steam per H.-P.
172.0	1,678							Upper figures shop only.
50.9	2,117	1,788	17,900	.....	17,900	.....	352 00	Lower figures shop and sawmill. Sawing ½ of time.
72.6	414	414	4,100	.....	4,100		39 00	
119.1	101	150	1,000	.....	1,500	.....	14 00	Sawing 2½ to 3¾ hours per day.
119.1	101	150	1,000	.....	1,500	.....	14 00	Not carded, taken same as above.
49.0	15,740	16,409	157,300	.....	164,000		919 00	
58.5	293	410	7,000	6,000	9,800	8,400	53 00	Fly-wheel pump.
61.6	265	371	6,400	5,500	8,900	7,600	53 00	"
57.7	421	585	10,100	8,000	14,000	12,000	74 00	Proper stroke, 12 in.
63.0	305	431	7,300	6,000	10,300	9,000	63 00	
45.1	2,607	2,607	26,100	.....	26,100		62 00	
708.	7,577	8,000	75,800	.....	80,000	.....	1,522 00	Calculated from heating of water, 225,700 lbs. per hour.
127.7	11,468	12,404	132,700	25,500	149,100	37,000	1,827 00	
49.1	3,240	3,691	77,800	77,800	88,600	88,600	355 00	
57.6	4,633	5,940	111,200	111,200	142,600	142,600	808 00	
63.3	1,247*	4,368	100	.....	105,800	105,800	262 00*	Extra pump ran 120 revs. during boiler test.
54.9	3,348*	4,464	8,000	8,000	107,100	107,100	518 00*	Avg. $\frac{6}{10}$ revs. per min. during boiler test.
56.0	2,192	5,634	52,600	52,600	135,200	135,200	364 00	
52.8	2,030*	3,045	100	.....	73,100	73,100	285 00*	Extra pump ran 120 revs. during boiler test.
59.4	3,697	5,789	88,700	88,700	138,900	138,900	688 00	
63.5	3,593	5,071	86,200	86,200	121,700	121,700	758 00	
56.5	2,680*	3,484	300	.....	83,600	83,600	446 00*	Extra pump ran 150 revs. during boiler test.
59.8	2,309	3,552	55,400	55,400	85,200	85,200	435 00	
56.5	3,564	5,580	85,500	85,500	133,900	133,900	592 00	
49.7	3,480	4,913	83,500	83,500	117,900	117,900	387 00	
39.4	3,668*	6,937	1,800	.....	166,500	166,500	.....	These two pumps are run alternately.
37.7	3,189	6,034	76,500	76,500	144,800	144,800	.....	No. 2 was running during boiler test.
52.8*	30,131*	68,502	727,700	725,400	1,644,900	1,644,900	4,411 00	Pumps not running not included in * totals.
57.7	92,163	142,119	1,474,000	954,100	2,514,700	1,885,100	11,452 00	Steam per H.-P. per hr. avg. of all.
65.6	793	991	19,000	16,000	23,800	19,000	180 00	These two pumps usually run alternately.
77.8	956	1,195	22,900	19,900	28,700	24,000	270 00	
63.0	80	80	500	500	500	500	.. ..	Runs ½ of time.
34.6	3,580	3,580	48,300	48,300	48,300	48,300	.....	Said to be avg. conditions. Avg. work 13½ hrs.
34.	2,720	2,720	36,700	36,700	36,700	36,700	.....	Not carded, data approximate, avg. work 13¼ hrs.
34.7	7,255	8,566	106,500	103,500	138,000	133,000	450 00	

Number of Card.	Description.	Colliery Number.	Location.	Steam						
				No. of Steam Cylinders.	Diameter Steam Cylinders.	Diameter Piston Rod.	Stroke.	Diameter Steam Pipe.	Steam Line.	Separator
				No.	In.	In.	In.	In.		
101	Fan engine	1	Short Mt.	1	24	$3\frac{5}{8}$	36	4	B	Y
100	"	2	"	1	24	$3\frac{5}{8}$	36	4	B & E	Y
102	"	....	East drift L. V.	1	12	$2\frac{1}{4}$	22	3	C	Y
	Fan engines, totals and averages	....		3	....	....	....	....		
142 & 143	Pair hoisting engines	....	Lykens Valley Slope	2	18	4	60	....	C	Y
105	Hoisting engine	....	Short Mt. Slope	1	27	4	48	....	B	Y
129, 130	Pair hoisting engines	....	No. 4 Slope	2	18	$2\frac{15}{16}$	36	....	M & N	Y
103	Accommodation engine	....	Bull Slope	1	12	$2\frac{3}{8}$	24	....	Boil. H.	N
	Hoisting engines, totals and averages	....		6	....	....	....	....		
132	Breaker engine	....	Lykens Valley Breaker	1	22	$3\frac{1}{4}$	48	....	A	Y
106	Separator engine	....	" Separator	1	16	$2\frac{5}{8}$	22	....	A	Y
104	Electric haulage engine	....	"	1	14	$3\frac{1}{4}$	48	....	B	..
138	Large shop engine	....	Blacksmith shop and saw mill	1	14	....	24	....	B	..
147	Small shop engine	....	" L. V.	1	$9\frac{1}{8}$	....	12	....	B	..
133	Prop saw oscillating engines	....	L. V. bank	2	$5\frac{1}{8}$	$\frac{5}{8}$	7	1	C	N
—	" " "	....	Short Mt. bank	2	$5\frac{1}{8}$	$\frac{5}{8}$	7	1	B	N
	Misc. outside engines, totals and avs.	....		9	....	....	....	....		
107	Boiler feed pump	....	Bull boiler house, end	1	8	$1\frac{1}{4}$	12	$1\frac{1}{2}$	Boil. H.	N
108	"	....	" side	1	8	$1\frac{1}{4}$	12	$1\frac{1}{2}$	Boil. H.	N
134, 135	Jeansville duplex boiler feed pump	....	"	2	10	$1\frac{1}{2}$	$10\frac{3}{4}$	....	Boil. H.	N
139, 140	" " " "	....	New boiler house	2	10	$1\frac{1}{2}$	$10\frac{1}{2}$	....	Boil. H.	N
141	Allison & Bannan pump	....	Mine water to breaker	1	25	$3\frac{1}{2}$	$104\frac{1}{2}$	....	D	
136	Pulsometers	....	L. V. Breaker	2	9	....	....	4	A	N
	Outside pumps, totals and averages	....		9	....	....	....	....		
131	Bull pump, single acting	1	Bull B. H. pumps, No. 1 Level	1	44	$6\frac{1}{2}$	116	7	Boil. H.	N
111, 112	Griscomb duplex pump	2	No. 1 Level	2	22	$3\frac{3}{8}$	36	5	K	
109	Carter & Allen pump	3	"	1	26	$4\frac{5}{8}$	67	3	K	
110	Allison pump	4	"	1	38	4	70	4	K	
123	"	1	No. 2 Level, Lyk. Vn.	1	36	4	70	4	I & J	
124	Allison & Bannan pump	2	" "	1	25	$3\frac{1}{2}$	$71\frac{1}{2}$	3	I & J	
114	Allison pump	3	" White's Vn.	1	38	4	70	4	F	
113	"	4	" "	1	30	$3\frac{3}{4}$	69	3	F	
125	Allison & Bannan pump	1	No. 3 Level	1	27	$3\frac{5}{8}$	$106\frac{1}{2}$	3	M	
126	Allison pump	2	"	1	27	$3\frac{1}{2}$	68	3	I & J	
127	"	3	"	1	38	4	$69\frac{1}{2}$	4	F	
128	"	4	"	1	30	4	$69\frac{1}{2}$	3	F	
119, 120	}									
121, 122	} Jeansville compound duplex pump	1	No. 4 Level	$4\frac{1}{2}$	24	$3\frac{1}{2}$	} 36	5	P & Q	
				$4\frac{1}{2}$	38	$4\frac{5}{8}$				
115, 116	}									
117, 118	} " " " "	2	"	$4\frac{1}{2}$	24	$3\frac{1}{2}$	} 36	5	P & Q	
				$4\frac{1}{2}$	38	$4\frac{5}{8}$				
	Mine pumps, totals and averages	....		21	....	....	....	....		
	Entire Colliery Plant, totals and avs.	....		48	....	....	....	....		
144	Blake pump	....	Fresh-water pump station	1	16	....	16	....		N
145	Cameron pump	....	" " "	1	18	....	$30\frac{1}{2}$	....		N
146	Boiler feed pump	....	" " "	1	7	....	8	....		N
137	Woodbury engine	....	Lykens electric plant	1	14	....	18	....		Y
—	Westinghouse engine	....	" "	2	$13\frac{1}{2}$	....	12	....		Y
	Fresh-water Pump. Sta. totals and avs.	....		6	....	....	....	....		

TABLE 5.—LYKENS VALLEY COAL COMPANY—ENGINES AND PUMI

	Steam Cylinders.				Water Data.				Hoisting Data.					Speed.		Power and Stea		
	Separator.	Trap.	Cocks Open = O.; or Closed = C.	Exhaust Into.	Diameter Plunger.	Diam. Column.	Length Column Pipe.	Lift, Including Vertical Suction.	Revolutions per Hoist.	Hoists per Day, Test.	Hoists per Day, Maximum.	Hoists per Idle Day.	Steam per Hoist.	Revolutions per Minute, Test.	Revolutions per Minute, Max.	M. E. P., Test.	Horse Power, Test.	Indicated Steam per H.-P. per
					In.	In	Ft.	Ft.	Revs.	No.	No.	No.	Lbs.	Revs.	Revs.	Lbs.	H.-P.	Lb
E	Yes	Yes	C.	Atmosphere	....	....	....	....	....	....	....	....	....	58	58	8.2	41.4	
	Yes	Yes	C.	"	....	....	....	....	....	....	....	....	....	60	60	8.15	41.2	
	Yes	Yes	C.	"	....	....	....	....	....	....	....	....	....	45	45	23.7	13.7	
	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	96.3	....
	Yes	Yes	¼ turn o.	Atmosphere	....	....	....	....	150	C. 50 A. 35	100 40	... 12	1,070 760	60 60	100 60	....	400.0 192.6	E. 47, V
	Yes	Yes	C.	"	....	....	....	....	170	210	250	10	807	64 130	{ 130	25.2	340.0	
N	Yes	Yes	¼ turn o.	Ditch	....	....	....	....	128	103	150	8	209	110	....	27.0	275.0	
H.	No	No	O.	Atmosphere	....	....	....	....	250	17	25	10	123	102	.....	16.4	32.9	
	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1,038.	.....
	Yes	Yes	C.	Atmosphere	....	....	....	....	.....	.....	..	....	.....	65	65	25.1	156.3	
	Yes	Yes	C.	"	....	....	....	....	.....	.....	.....	....	.....	84	84	42.5	79.7	
	....	....	O.	"	....	....	....	....	.....	.....	.....	....	.....	88	88	14.0	44.2	
	....	....	.....	"	....	....	....	....	.....	.....	.....	....	.....	120	.....	30.1	23.3	
	....	....	.....	"	....	....	....	....	.....	.....	.....	....	.....	80	.....	27.1	41.6	
	....	....	.....	"	....	....	....	....	.....	.....	.....	....	.....	130	.....	11.1	5.7	
	No	No	O.	"	....	....	....	....	.....	.....	.....	....	.....	160	.....	12.5	3.4	1
	No	No	O.	"	....	....	....	....	.....	.....	.....	....	.....	160	.....	12.5	3.4	1
	....	....	.....	.....	....	....	....	....	.....	.....	.....	....	.....	....	....	....	321.	.....
H.	No	No	C.	Heater	4	....	....	....	.....	.....	.....	....	.....	50	70	23.1	5.0	
H.	No	No	C.	"	4	....	....	....	.....	.....	.....	....	.....	50	70	28.5	4.3	
H.	No	No	¼ turn o.	"	6	....	....	....	.....	.....	.....	....	.....	18	25	48.9	7.3	
H.	No	No	.....	Atmosphere	6	....	....	....	.....	.....	.....	....	.....	17.7	25	33.9	4.8	
	....	....	.....	Heater	12	....	....	....	.....	.....	.....	....	.....	7	.....	18.8	57.2	
	No	No	.....	Column	....	4	....	94	.....	.....	.....	....	.....	.....	.....	.....	10.7	.....
	....	....	.....	.....	....	....	....	....	.....	.....	.....	....	.....	....	....	....	89.8	.....
H.	No	No	⅛ turn o.	Heater	21⅛	20	550	319	.....	.....	.....	....	.....	4.17	4½	37.6	68.4	4
			⅛ turn o.	Syphon condenser	9 5/16	10	541	{ 318	.....	.....	.....	....	.....	19.5	25	31.1	80.5	5
			"	Tank	11 5/16	8	568	293	.....	.....	.....	....	.....	4	14	25.5	19.7*	3
J			"	Tail pipe	16½	14	520	317	.....	.....	.....	....	.....	6	8	25.1	61.0*	5
J			"	Syphon condenser	16½	12	489	331	.....	.....	.....	....	.....	3½	9	31.5	39.9	5
			"	Tank	12½	8	485	331	.....	.....	.....	....	.....	6	9	26.6	39.2*	4
			"	Syphon condenser	16¼	12	420	328	.....	.....	.....	....	.....	5¼	9	27.0	62.2	5
			"	"	12⅜	8 { 12 {	492	328	.....	.....	.....	....	.....	8.5	12	27.2	56.6	5
J			"	Ditch	12⅛	8	325	321	.....	.....	.....	....	.....	5	6½	30.8	47.4*	5
			"	"	12⅝	10	325	321	.....	.....	.....	....	.....	6.5	10	30.1	38.6	5
			"	Syphon condenser	16	12	342	326	.....	.....	.....	....	.....	5¼	9	27.4	63.0	5
			"	"	14¼	10	358	326	.....	.....	.....	....	.....	8.5	12	33.0	69.9	4
Q			"	Tail pipe	10	10	800	673	.....	.....	.....	....	.....	9½	17½	26.75 12.30	46.0* 46.9*	{ {
Q			"	Syphon condenser	10	10	800	660	.....	.....	.....	....	.....	9¼	17½	26.8 10.1	46.1 38.5	{ {
	....	....	.....	.....	....	....	....	....	.....	.....	.....	....	.....	....	....	....	569.8*	.....
	....	....	.....	.....	....	....	....	....	.....	.....	.....	....	.....	....	....	....	2,114.9	.....

Separator & trap on each level

LYKENS VALLEY COAL COMPANY—FRESH-WATER PUMPING STATION.

....	No	No	.....	Atmosphere	8	....	....	.....	.....	.....	.....	....	.....	30	37½	25.2	12.1	6
....	No	No	.....	"	8	....	....	.....	.....	.....	.....	....	.....	16	20	20.1	12.3	3
....	No	No	.....	"	4½	....	....	.....	.....	.....	.....	....	.....	25	.....	33.5	1.3	5
....	Yes	Yes	.....	"	....	....	....	.....	.....	.....	.....	....	.....	246	246	30.1	103.5	4
....	Yes	Yes	.....	"	....	....	....	.....	.....	.....	.....	....	.....	300	300	.....	80.0	.....
....	....	....	.....	.....	....	....	....	.....	.....	.....	.....	....	.....	.....	.....	.....	197.0	.....

TABLE 6.—SUMMIT BRANCH COAL COMPANY—ENGINES AND PUMPS, APRIL AND MAY, 1898.

Number of Card.	Description.	Colliery Number.	Location.	Steam Cylinders. No. of Steam Cylinders.	Diameter Steam Cylinders.	Diameter Piston Rod.	Stroke.	Diameter Steam Pipe.	Steam Line.	Separator.	Trap.	Cocks Open=O.; or Closed=C.	Exhaust Into.	Water Data. Diameter Plunger.	Diameter Column.	Length Column Pipe.	Vertical Lift, including Suction.	Hoisting Data. Revolutions per Hoist.	Hoists per Day, Test.	Hoists per Day, Maximum.	Hoists per Idle Day.	Steam per Hoist.	Power & Steam Economy. Revolutions per Minute, Test.	Revolutions per Minute, Maximum.	M. E. P., Test.	Horse Power, Test.	Ind. Steam per H. P. per Hour, Test.	Total Steam per H.P. per Hour, incl. Cylinder Condensation.	Total Steam per Hour, Test.	Total Steam per Hour, Maximum.
				No.	In.	In.	In.	In.						In.	In.	Ft.	Ft.	Revs.	No.	No.	No.	Lbs.	Revs.	Revs.	Lbs.	H.P.	Lbs.	Lbs.	Lbs.	Lbs.

6.—SUMMIT BRANCH COAL COMPANY—ENGINES AND PUMPS, APRIL AND MA

Steam Cylinders.					Water Data.				Hoisting Data.					Speed.		Power & Steam Economy.			
Pipe.	Steam Line.	Separator.	Trap.	Cocks Open=O.; or Closed=C.	Exhaust Into.	Diameter Plunger.	Diameter Column.	Length Column Pipe.	Vertical Lift, including Suction.	Revolutions per Hoist.	Hoists per Day, Test.	Hoists per Day, Maximum.	Hoists per Idle Day.	Steam per Hoist.	Revolutions per Minute, Test.	Revolutions per Minute, Maximum.	M. E. P., Test.	Horse-Power, Test.	Ind. Steam per H.-P. per Hour, Test.	Total Steam per H.P. per Hour, Incl. Cylinder Condensation.
n.						In.	In.	Ft.	Ft.	Revs.	No.	No.	No.	Lbs.	Revs.	Revs.	Lbs.	H.-P.	Lbs.	Lbs.
														WILLIAMSTOWN PLANT.						
4	E	Yes	Yes	C.	Atmos.	....	....	.....	.....	.....	.....	.....	....	.....	64	.....	10.75	55.2	65.7	75.5
4	E	No	No	C.	"	....	....	.....	.....	.....	.....	.....	....	.....	60	.....	7.95	39.3	77.0	88.5
..	C	"	"	½ turn o.	.....	....	....	.....	.....	.....	.....	.....	....	.....	42	.....	13.1	10.1	63.4	72.9
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	104.6	.....	80.2
..	C	No	No	¼ turn o.	Syp. con.	....	....	.....	.....	108	C. 110	150	20	1,078	64	.....	40.6	658.0	49.2	53.1
..	B	"	"	O.	.....	....	....	.....	.....	274	A. 20	20		270	100	.....	27.2	*93.0	*50.0	*55.0
..	.....	.....	.....	.....	.....	....	....	.....	.....	.....	C. 150	200	....	.....	100	.....	27.2	106.4	44.3	47.8
..	Boil. H.	No	No	¼ turn o.	Atmos.	....	....	.....	.....	114½	A. 15	15	15	225	120	.....	27.9	62.2	28.4	32.7
4	A	Yes	Yes	O.	"	....	....	.....	.....	450	98	120	....	18.9	200	.....	29.4	17.4	33.5	38.5
3	G	No	No	O.	"	....	....	.....	.....	.....	67	90	....	30.5	246	.....	14.1	5.0	133.5	153.5
														Empty	50	.....	4.5			
														Hoisting ...	136	.....	14.4	46.3	80.0	92.0
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	855.9	.....	51.6
2	A	Yes	Yes	Autom.	At. & heat	....	....	.....	.....	.....	.....	.....	....	.....	146	146	24.8	145.0	33.5	38.5
2	G	No	No	O.	Atmos.	....	....	.....	.....	.....	.....	.....	....	.....	140	140	10.2	12.8	63.7	73.2
2	F	"	"	½ turn o.	"	....	....	.....	.....	.....	.....	.....	....	.....	140	140	22.5	7.2	31.8	35.0
..	F	Yes	Yes	C.	"	....	....	.....	.....	.....	.....	.....	....	.....	50	Lgt ..	11.6	13.7	65.4	72.0
																Saw..	31.4	36.7	45.2	49.0
..	Boil. H.	No	No	¼ turn o.	Heater	....	....	.....	.....	Slant conveyer only.					78	.....	10.6	*15.2	66.8	76.8
..	"	.....	.....	.....	.....	....	....	.....	.....	Monobar conveyer only.					78	.....	12.5	19.6	61.0	70.2
..	"	.....	.....	.....	.....	....	....	.....	.....	Both convs. estimated.					78	.....	.....	*30.0	.....	65.0
..	C	No	No	½ turn o.	.....	....	....	.....	.....	.....	.....	.....	....	.....	200	.....	4.3	2.6	183.0	201.0
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	206.7	.....	47.7
..	Boil. H.	No	No	½ turn o.	Atmos	6	....	.....	.....	.....	.....	.....	....	.....	15.87	21	48.6	7.3	48.0	52.0
..	"	"	"	½ turn o.	Heater	6	....	.....	.....	.....	.....	.....	....	.....	20.33	30	29.1	12.0	57.5	63.3
..	"	"	"	"	"	4	....	.....	.....	.....	.....	.....	....	.....	24	30	21.9	3.7	62.5	67.5
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	23.0	.....	60.4
4	B, E, & H	No	No	C.	Tail pipe	10	10	648	241	.....	.....	.....	....	.....	11.6	26	24.0	25.7	59.0	63.6
5	"	Yes	"	C.	"	12	10	648	242	.....	.....	.....	....	.....	11.4	15	27.2	53.9	59.5	64.3
5	"	No	"	C.	"	10	10	648	242	.....	.....	.....	....	.....	24.4	37.5	26.2	88.4	63.9	69.0
4	J	"	"	C.	"	10	....	.....	.....	.....	.....	.....	....	.....	16.4	25.0	28.1	31.9	52.4	56.6
4	B	Yes	"	C.	"	10	10	920	389	.....	.....	.....	....	.....	13.9	15.0	18.75	68.9	65.0	70.2
4	J	"	"	C.	"	9	8	1,004	389	.....	.....	.....	....	.....	20	28	20.4	48.0	42.6	46.0
5	J	"	"	C.	"	16	12	888	389	.....	.....	.....	....	.....	4.2	8.5	31.1	45.7	57.3	61.9
4	J	"	"	C.	"	10	10	983	389	.....	.....	.....	....	.....	17.8	25	34.4	43.7	47.8	51.6
4	L	"	"	C.	"	8	8	612	386	.....	.....	.....	....	.....	23.05	27.5	23.1	49.2	63.9	69.0
5	L	"	"	C.	"	10	12	638	386	.....	.....	.....	....	.....	8.8	18	26.6	32.4	47.3	51.1
5	M	"	"	C.	"	16	12	584	388	.....	.....	.....	....	.....	3.65	7	31.1	46.9	48.2	52.0
..	M	"	"	½ turn o.	"	10	8	600	386	.....	.....	.....	....	.....	32.2	50	31.8	56.2	56.6	60.1
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	590.9	.....	61.2
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1,781	.....	56.1
														BEAR VALLEY SLOPE PLANT.						
..	O	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	....	.....	50	.....	14.1	7.3	*72.7	80.0
..	O	Yes	Yes	½ turn o.	Atmos.	....	....	.....	.....	181	C. 20	160	....	245	200	.....	18.3	68.2	86.1	94.7
..	O	"	"	"	"	....	....	.....	.....	181	A. 19	20	20	160	200	.....	21.1	83.9	75.0	82.5
..	O	No	No	½ turn o.	Tail pipe	9	10	358	232	.....	.....	.....	....	.....	28	40	14.5	16.0	96.0	103.7
..	O	"	"	"	"	9	8	358	231	.....	.....	.....	....	.....	22.8	40	15.5	14.9	81.5	88.0
..	O	Yes	Yes	"	"	7	....	.....	.....	.....	.....	.....	....	.....	20.4	30	43.9	21.5	54.5	58.9
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	211.8	.....	85.9
														BEAR VALLEY SHAFT PLANT.						
..	Q	No	No	Autom.	Atmos.	....	....	.....	.....	.....	.....	.....	....	.....	52	85	16.5	40.2	24.3	39.2
..	P	"	"	½ turn o.	"	....	....	.....	.....	24	C. 8	400	....	212	42	.....	66.7	620.0	36.0	39.6
..	Q	"	"	O.	"	....	....	.....	.....	241	A. 51	.....	30	56	42	.....	9.1	*86.0	73.0	80.0
..	Boil. H.	.....	.....	.....	Heater	5	....	.....	.....	.....	8	60	...	40	132	.....	30.8	92.6	41.5	45.7
3	Q	No	No	O.	Tail pipe	8	8	.....	.....	.....	.....	.....	....	.....	28	.....	31.6	4.1	56.0	60.5
3	Q	"	"	O.	"	8	8	.....	.....	.....	.....	.....	....	.....	16.8	40	19.7	9.9	76.0	82.0
3	R & S	Yes	"	½ turn o.	Atmos.	12	10	472	338	.....	.....	.....	....	.....	14.5	40	36.0	15.5	60.4	65.2
	"	No	"	"	.....	9	6	488	342	.....	.....	.....	....	.....	13.5	30	38.5	25.4	66.6	71.9
	"	"	"	O.	Tail pipe	12	10	496	339	.....	.....	.....	....	.....	28	40	24.3	28.6	73.5	79.4
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	18	30	35.7	52.8	46.5	50.3
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	889.1	.....	44.1
														GREENFIELD PUMPING PLANT.						
..	.....	No	No	.....	Atmos.	7⅞	....	.....	.....	.....	.....	.....	....	.....	96	120	36.9	38.2	21.1	27.7
..	.....	"	"	O.	"	8	...	.....	.....	.....	.....	.....	....	.....	36	40	48.4	84.7	52.5	56.3	60.8
	.....	.....	.....	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	120.5	.....	28.4
														BIG LICK SLOPE.						
8	.....	Yes	Yes	C.	Atmos.	....	..	..	....	1st lift	34½	40	C. 100	...	C. 211	A. 149	.....	.....	.....	.....
										2d "	79½	30	A. 10	10	C. 487	A. 343	C. 76	96	41.5	512.6	33.6	39.1
										3d "	102½	4	... ..	....	C. 623	A. 444	A. 42	42	15.5	109.4	49.4	57.5
	.....	.....	.....	.....	.....	....	....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	3,515.	.....	51.5

Total H.-P. and average steam per H.-P. calculated, assuming all engines to run constantly.

MAY. 1898.

omy.	Steam Consumption.							Remarks.
incl. Cylinder Condensation.	Total Steam per Hour, Test.	Total Steam per Hour, Maximum.	Total Steam per Working Day, Test.	Total Steam per Idle Day, Test.	Total Steam per Working Day, Maximum.	Total Steam per Idle Day, Maximum.	Cost per Year of Steam Used Above 40 Lbs. per I. H.-P. per Hour.	
bs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	$	
.5	4,177	4,177	100,100	100,100	100,100	100,100	964 00	Mullen engine, 4½" governor, 25' fan.
.5	3,480	3,480	83,500	83,500	83,500	83,500	988 00	" " "
.9	736	736	17,700	17,700	17,700	17,700	172 00	Allison & Bannan engine.
.2	8,393	8,393	201,300	201,300	201,300	201,300	2,124 00	
.1	9,903	12,950	128,700	16,000	168,400	20,000	445 00	Hoists four cars. Single slope steam per rd. trip.
.0	*3,001	*4,000	39,000	4,000	*52,000	*4,000	.....	Valves badly out.
.8	2,518	3,357	*32,700	*3,300	43,600	3,300	81 00	Valves reset.
.7	173	213	1,800	.....	2,300	.....	.....	Third notch ; 36.8 ind. steam per H.-P. hr. in 2d notch
.5	123	165	2,000	.....	2,700			
.5 .0	1,350	1,350	.....	13,500	.....	13,500	159 00	Runs constantly ; hoists about ⅛ of time.
.6	14,067	18,035	171,500	33,500	217,000	36,800	685 00	
.5	5,562	6,192	55,600	.....	61,900	.....	.....	[30.7 and 35.3 per H.-P. With steam heat on 4⅓ lbs. per H.-P. extra ; 4th notch
.2	937	937	.....	9,400	.....	9,400	84 00	
.0	252	252	2,500	.....	2,500	.....	.....	Valve reset ; steam consumption reduced from 46.9 lbs.
.0 .0	1,196	1,196	12,000	.....	12,000	.....	65 00	Sawing ¼ of time ; valve reset from 91 lbs. steam H.-P.
.8	*1,167	*1,167	.....	11,700				
.2	1,376	1,376	.....	.....	13,800	.....	118 00	
.0	*1,950	*1,950	.....	.....	.....	19,500		
.0	523	523	6,800	.....	6,800	.....	83 00	Car on ¹⁄₁₀ of time, estimated.
.7	9,846	10,476	76,900	21,100	97,000	28,900	350 00	* Not included in averages.
.0	378	504	9,000	8,000	12,100	10,000	45 00	
.3	760	1,013	18,200	16,000	24,300	21,000	144 00	
.5	250	312	6,000	6,000	7,500	7,500	51 00	
.4	1,388	1,829	33,300	30,000	43,900	38,500	240 00	
.6	1,635	3,682	39,200	39,200	88,400	88,400	304 00	
.3	3,467	4,562	83,200	83,200	109,500	109,500	679 00	
.0	6,097	9,370	146,300	146,300	224,900	224,900	1,328 00	
.6	1,827	2,785	43,900	43,900	66,900	66,900	275 00	One-half of duplex pump, Nos. 17 and 19.
.2	4,896	5,283	117,500	117,500	126,800	126,800	1,078 00	
.0	2,207	3,090	53,000	53,000	74,200	74,200	149 00	
.9	2,829	5,725	67,900	67,900	137,400	137,400	518 00	
.6	2,255	3,167	54,100	54,100	76,000	76,000	260 00	One-half of duplex pump cards 17 and 19.
.0	3,399	4,021	81,600	81,600	96,500	96,500	739 00	
.1	1,656	3,387	39,700	39,700	81,300	81,300	187 00	Cameron steam, Allison water end.
.0	2,443	4,690	58,600	58,600	112,600	112,600	292 00	
.1	3,438	5,335	82,500	82,500	128,000	128,000	585 00	
.2	36,149	55,097	867,500	867,500	1,322,500	1,322,500	6,394 00	
.1	69,843	93,830	1,350,500	1,153,400	1,881,700	1,628,000	9,793 00	
.0	584	584	14,000	14,000	14,000	14,000	143 00	
.7	410	1,435	8,500	4,000	19,600	4,000	227 00	These engines are to be replaced soon. In very poor condition.
.5	437	1,529						
.7	1,659	2,370	39,900	39,900	56,900	56,900	489 00	
.0	1,323	2,331	31,700	31,700	55,900	55,900	301 00	
.9	1,266	1,862	30,400	30,400	44,700	44,700	199 00	
.9	5,679	10,111	124,500	120,000	191,100	175,500	1,369 00	
.2	1,576	4,956	37,800	37,800	119,000	119,000	.....	[steam. Little work ; 5% cut off ; full work est. 177 H.-P., 28 lbs.
.6	78	8,480	4,900	4,900	84,800	5,600		
.0	125							
.7	14	240	300	300	2,400	2,400		
.5	256	350	6,100	6,100	8,400	8,000	41 00	
.0	812	1,933	19,500	19,500	46,400	46,400	204 00	
.2	1,010	2,786	24,200	24,200	66,900	66,900	192 00	
.9	1,827	4,060	43,800	43,800	97,400	97,400	397 00	
.4	2,272	3,246	54,500	54,500	77,900	77,900	552 00	
.3	2,634	4,360	63,200	63,200	105,400	105,400	267 00	
.1	10,604	30,441	254,300	254,300	608,600	529,000	1,653 00	
.7	3,421	4,276	80,400	74,000	100,500	92,000	.....	With cross-pipe closed 19.9 ind. steam per H.-P.
.8	3,191	3,546	1,600	1,600	1,800	1,800	.....	Extra pump runs about one-half hour per day.
.4	3,416	4,262	82,000	75,600	102,300	93,800	.....	Riedler 23½ Cameron one-half hour in average & totals.
...								
9.1	2,614	4,708	26,100	3,400	47,100	3,400	.....	C = coal ; A = accommodation.
7.5								
1.5	92,156	143,352	1,837,400	1,606,700	2,830,800	2,429,700	12,815 00	

TABLE 9.—STEAM LINES AND CONDENSATION LOSSES—LYKENS VALLEY COAL COMPANY.

Letter.	Location.	Condition.	Covering.	Steam Pressure.	Inside = I. Outside = O.	Size and Length of Pipe.																Condensing Surface.	Condensation.				Remarks.
						30" Wt.	24" Wt.	16" Cast.	12" Cast.	10" Wt.	8" Cast.	8" Wt.	6" Cast.	6" Wt.	5" Cast.	5" Wt.	4" Wt.	3" Wt.	2" Wt.	1¼" Wt.	1" Wt.		Winter. Temp. Outs., 30°; Inside, 70°. Heating Buildings, Winter Only.	Winter. Steam Lines.	Winter. Total Winter Condens., 24 Hrs.	Summer. Outs., 70°; Ins., 90°. Total Summer Condens., 24 Hrs.	
						Ft.	Ft.	Ft	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Sq. Ft.	Lbs.	Lbs.	Lbs.	Lbs.	
—	Lines in boiler houses	Good	Good	80 / 100	O.	....	....	...	600	....	....	....	....	.....	600	.....	....	....	....	....	....	3,468	.....	12,346	12,346	10,473	Est. equiv.
A	To breaker	Good	Good	80	O.	....	....	....	....	....	....	....	200	1,560	....	10	80	....	....	....	....	3,413	2,300	11,706	14,006	10,068	Breaker and separator.
B	Short Mt. Bank	Good	Good	80	O.	....	....	....	....	....	840	....	350	.....	150	30	400	....	....	....	200	3,971	21,100	13,621	34,721	11,714	Heats shop, shifting house, and electric plant.
C	Lykens Valley Bank	Good	Good	80	O.	....	....	....	....	....	....	....	220	.....	....	....	....	....	300	380	....	846	9,100	2,902	12,000	2,496	Heats shifting house and supply house.
D	To mine water pump	Good	Good	80	O.	....	....	....	....	....	....	....	....	....	....	....	....	100	20	....	..	108		370	370	319	
E	Bal. boilers to No. 1 level	Good	Good	80	O.	....	....	....	....	....	....	....	375	.....	....	....	35	....	....	....	....	813		2,789	2,789	2,398	
					I.	....	....	....	....	....	....	....	.....	558	....	....	....	....	....	....	....	1,032		3,044	3,044	2,797	
F	New B. H. White's Vn	Good	Good	80	O.	....	....	....	....	....	....	....	210	.....	....	....	80	100	....	....	....	636		2,181	2,181	1,876	
					I.	....	....	....	....	....	....	....	1,610	.....	350	....	....	....	....	....	....	4,020		11,859	11,859	10,894	
G	High press. to No. 4 slope	Out for repairs.		100	O.	...	....	....	....	....	....	250	.....	.....	.....	.....	.....	.....	.....	.....	....	625		2,313	2,313	1,938	No steam on.
					I.	....	....	....	....	....	....	1,730	.....	.....	.....	.....	.....	.....	.....	.....	...	4,325		13,408	13,408	12,369	No steam on.
H	From line B to No. 1 level	Good	Good	80	O.	....	....	....	....	....	....	....	30	.....	.....	.....	.....	.....	.....	.....	..	63		216	216	186	
					I.	....	....	....	....	...	....	....	612	...	.....	.....	...	....	.....	.....	....	1,291		3,808	3,808	3,499	
I	No. 1 level to No. 3 level	Good	Good	80	I.	....	....	..	....	....	....	....	930	.....	.....	.....	.....	75	.....	.....	....	2,161		6,375	6,375	5,856	
J	No. 1 level to No. 3 level	Good	Good	80	I.	....	....	....	....	....	....	....	.....	.....	.....	975	40	25	.....	.....	....	1,731		5,106	5,106	4,691	
K	On No. 1 level	Good	Good	80	I.	....	....	....	....	....	....	....	.....	.....	400	200	....	....	.....	.....	....	1,052		3,103	3,103	2,851	
L	No. 3 level, White's to Lyk. Vn.	Good	Good	80	I.	....	....	....	....	....	....	....	.....	.....	.....	260	....	....	.....	.....	....	442		1,304	1,304	1,198	
M	No. 3 level, Lyk. Vn. to No. 4 sl.	Good	Good	80	I.	....	....	....	....	....	....	....	.....	.....	.....	640	....	....	.....	.....	....	1,088		3,210	3,210	2,948	
N	No. 3 lev., White Vn. to No. 4 sl. engs.	Good	Good	80	I.	....	....	....	....	....	....	....	.....	525	.....	.....	.....	.....	.....	.....	....	971		2,864	2,864	2,631	
O	No. 3 level, G to No. 4 sl. engs.	Good	Good	100	I.	....	....	....	....	....	....	....	.....	332	.....	.....	.....	.....	.....	.....	....	614		1,903	1,903	1,756	
P	No. 3 level to No. 4 level pumps	Good	Good	100	I.	....	....	....	....	....	....	....	.....	.....	.....	965	.....	.....	.....	.....	....	1,641		5,087	5,087	4,693	
Q	No. 3 level to No. 4 level pumps	Good	Good	100	I.	....	....	....	....	....	....	....	.....	795	.....	120	.....	.....	.....	.....	....	1,675		5,192	5,192	4,890	No steam on.
	Receivers	Good	Good	80 / 100	O.	4 = 60 ft. 3¼"	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	552		1,987	1,987	1,657	
					I.	8 = 120 ft. 3¼"	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1,104		3,257	3,257	2,992	
	Grand total					....	....	....	600	....	840	1,980	4,537	3,770	1,500	3,200	635	300	320	380	200	37,642	32,500	118,951	151,451	107,200	
	During test of boilers lines Q and G off. No heating. Temp. 60°.																									81,725	
	Outside area of pipe incl. flanges, per 100'					850	700	530	400	315	280	250	211	185	178	170	123	95	64	50	35	All sq. ft. per 100 ft. length.					

Steam line A—103 sq. ft. heating pipe.
B—926
C—400

See Map A for Outside Lines and B for Inside Lines.

0.012 lb. condensation in 24 hrs. per sq. ft. per degree difference in temperature. Condensation calculated on pressures 70 lbs. and 100 lbs. average.

TABLE 10.—STEAM LINES AND CONDENSATIO

Letter.	Location.	Condition.	Covering.	Steam Pressure.	Inside = I. Outside = O.	Size and 30" Wt.	24" Wt.	16" Cast.	12" Cast.	10" Wt.	8" Cast.	8" Wt.	6" Cast.
	WILLIAMSTOWN PLANT.—See Maps C for outside, and D for inside lines.												
						Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.
—	Boiler house	Fair	Good	80 / 180	O.	104	70	60	....	....	....	....	200
A	No. 2 breaker and plane	Good	Good	80	O.	....	....	....	....	810	....	....	....
B	High pres. to foot No. 3 Sl.	Bad	....	120	O.	....	....	....	....	....	....	530	....
					I.	....	....	....	....	....	....	3,050	....
C	Low pres. E. side tunnel	Poor	Fair	80	O.	....	....	....	....	....	....	220	....
					I.	....	....	....	....	....	....	1,340	....
D	Low pres. W. side tunnel	Poor	Fair	80	O.	....	....	....	....	....	....	400	....
					I.	....	....	....	....	....	....	1,100	....
E	Low pres. No. 1 fan to ft. No. 3 Sl.	Poor to bad	Fair	80	O.	....	....	....	....	....	....	675	50
					I.	....	....	....	....	....	....	....	....
F	Shop and barn	Good	Good	80	O.	....	....	....	....	....	....	....	....
G	Separator	Poor	4" poor / 3" none	80	O.	....	....	....	....	....	....	....	....
H	Tunnel lev. to No. 3 Sl. bot.	Unknown	....	80	I.	....	....	....	....	....	....	....	....
I	Tunnel lev. to No. 2 pocket	Fair	Good	80	I.	....	....	....	....	....	....	....	....
J	No. 3 pocket to No. 3 Sl. bot.	Good	Good	80	I.	....	....	....	....	....	....	....	....
K	Abv. No. 3 pocket to No. 3 Sl. bot.	....	....	....	I.	....	....	....	....	....	....	....	....
L	W. of No. 4 Slope	Good	Good	120	I.	....	....	....	....	....	....	....	....
M	E. of No. 4 Slope	Poor	Fair	120	I.	....	....	....	....	....	....	....	....
N	Whites Vn. to ft. No. 4 Sl.	Upper poor. Lower unkno'n.	Good	120	I.	....	....	....	....	....	....	....	....
	Totals, Williamstown Plant	....	....	....	.....	104	70	60	....	810	....	7,335	250
	BEAR VALLEY SLOPE PLANT.—See Map E.												
O	B. V. Slope, fan and engs.	Good	Good	80	O.	....	....	....	....	....	....	....	....
					I.	....	....	....	....	....	....	....	....
	Totals, Bear Valley Slope Plant	....	....	....	.....	....	....	....	....	....	....	....	....
	BEAR VALLEY SHAFT PLANT.—See Map E.												
—	Boiler house	Good	Good	80 / 100	O.	....	....	....	....	....	....	....	40
P	Hoisting engines	Good	Good	100	O.	....	....	....	....	....	....	....	....
Q	Fan, shop, and shaft pumps	Good	Good	80	O.	....	....	....	....	....	....	150	....
		Fair	Wet	80	I.	....	....	....	....	....	....	....	....
R	Summit Slope line	Good	Good	80	O.	....	....	....	....	....	....	....	....
		Fair	Good	....	I.	....	....	....	....	....	....	....	....
S	Summit Slope	Fair	Poor	80	O.	....	....	....	....	....	....	....	....
		Fair	Fair	....	I.	....	....	....	....	....	....	....	....
	Totals, Bear Valley Shaft Plant	....	....	....	.....	....	....	....	....	....	....	150	40

Condensation calculated on pressure of 70 and 100 lbs. averages. 0.012 lb. co

ATION LOSSES—SUMMIT BRANCH COAL COMPANY.

AND LENGTH OF PIPE.									Condensing Surface.	CONDENSATION.			REMARKS.
										WINTER. Temp. Outs., 30°; Inside, 70°.		SUMMER. Outs., 70°; Ins., 86°.	
6" Cast.	6" Wt.	5" Cast.	5" Wt.	4½" Cast.	4" Wt. & C.	3" Wt.	2" Wt.	1", 1¼" & 1½" Wt.		Heating Buildings, Winter Only.	Steam Lines.	Total Winter, Condens., 24 Hrs.	Total Summer Condens., 24 Hrs.	
Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Sq. ft.	Lbs. Supply B. H. Br. E. H.	Lbs.	Lbs.	Lbs.	Est. equiv.
200	.....	.....	.....	.....	1,030	.....	.....	.....	3,504	2,200	12,474	14,674	10,617	{ Incl. temp. 4°. { To coal plane.
.....	60	.....	.....	.....	400	.....	.....	.....	3,153	1,440	10,815	12,255	9,300	Breaker heated by exhaust steam.
.....	.....	.....	.....	.....	.....	.....	.....	.....	1,375	.....	5,088	5,088	4,263	
.....	120	.....	.....	.....	15	.....	.....	.....	6,493	.....	20,128	20,128	18,570	
.....	.....	.....	.....	.....	.....	.....	.....	.....	550	.....	1,887	1,887	1,622	
.....	.....	.....	180	.....	.....	.....	100	.....	3,720	.....	10,974	10,974	10,081	
.....	.....	.....	.....	.....	.....	.....	.....	.....	1,000	.....	3,430	3,430	2,950	
.....	.....	.....	.....	.....	.....	.....	.....	.....	2,750	.....	8,112	8,112	7,453	
50	.....	230	.....	70	.....	.....	.....	.....	2,307	.....	7,913	7,913	6,806	
.....	2,510	.....	.....	.....	.....	.....	.....	.....	4,644	.....	13,700	13,700	12,585	
.....	.....	.....	.....	.....	.....	450	300	½ 1½" ½ 20 ½	630	11,970	2,161	14,131	1,858	[offices. Heat to shop, barn and
.....	.....	.....	.....	.....	425	850	.....	.....	1,331	.....	25,900	25,900	22,150	{ 4" Dbl. cond. { 3" Bare 8 times cond.
.....	1,850	.....	.....	.....	.....	.....	.....	.....	3,423	.....	10,098	10,098	9,276	
.....	1,070	20	.....	.....	20	.....	.....	.....	2,041	.....	6,021	6,021	5,531	
.....	1,670	80	.....	.....	40	170	.....	.....	3,461	.....	10,210	10,210	9,479	
.....	620	.....	.....	.....	.....	.....	.....	.....	(1,147)	.....	.....	.....	.....	Not in use.
.....	.....	.....	950	.....	20	.....	.....	.....	1,640	.....	5,084	5,084	4,700	
.....	.....	.....	850	.....	.....	.....	.....	.....	1,445	.....	4,480	4,480	4,123	
.....	915	.....	30	.....	.....	.....	.....	.....	1,744	.....	5,406	5,406	4,988	
250	8,815	330	2,010	70	2,940	1,470	400	20	45,211	15,610	163,881	179,491	146,272	

.....	440	.....	.....	.....	.....	50	.....	¼ 1½" ¼ 180 ¼	937	2,200	3,215	5,415	2,764	
.....	650	.....	.....	.....	100	20	.....	.....	1,392	.....	4,106	4,106	3,772	
.....	1,090	.....	.....	.....	100	70	.....	180	2,329	2,200	7,321	9,521	6,536	

40	.....	40	.....	..	80	100	.....	.....	349	.....	1,197	1,197	1,030	
.....	.....	.....	.....	.....	.....	.....	.....	.....	375	1,400	1,388	2,788	1,163	
.....	.....	.....	300	...	30	15	.....	¼ 1" ¼ 150 ¼	625	2,400	2,154	4,554	1,844	
.....	.....	.....	.....	.....	830	50	.....	.....	1,021	.....	6,024	6,024	5,513	Dbl. cond. for wet shaft.
.....	.....	.....	1,550	.....	.....	.....	.....	.....	2,635	.....	9,038	9,038	7,773	
.....	.....	.....	.....	500	...	40	.....	.....	788	.....	2,325	2,325	2,135	
.....	.....	.....	.....	40	.....	100	.....	.....	155	.....	1,063	1,063	915	Dbl. cond. Poor cover'g.
.....	.....	.....	.....	...	.....	500	.....	.....	475	.....	1,401	1,401	1,287	
40	.....	40	1,850	540	1,140	805	.....	150	6,423	3,800	24,530	28,390	21,660	

lb. condensation in 24 hours per sq. ft. surface per degree difference in temperature.

UNIV. OF MICH.

TABLE 12.—FRICTION AND CONDENSATION I

Loss of pressure by friction, $p_1 - p_2 = \frac{Q^2 WL}{c^2 d^5}$ (Kent, p. 671).

Q = Quantity in cubic feet per minute.
W = Weight steam per cubic foot.
L = Length pipe in feet = 1,000.

c = Varying constant, see table.
d = Diameter pipe in inches.

W @ 80 lbs. =
W @ 100 lbs. =
W @ 120 lbs. =

Size Pipe.	"C"	$c^2 d^5$	Quantity per Hour.	1,000 Lbs.			2,500 Lbs.			5,000	
			Cubic feet per minute	76.66	63.83	55.0	191.6	159.6	137.5	383.3	319.
			Gauge pressure, lbs	80	100	120	80	100	120	80	10
			Loss pressure friction	14.4	11.8	8.2	.....	75.2	64.7		
			" " condensation	0.6	0.7	0.9	.....	0.1	0.1		
2"	52.7	88,800	" " total	15.0	12.5	9.1	.....	75.3	64.8		
			Condensation per hour	84	89	93	.....	89	93		
			Equivalent condensation due to friction	4	3	2	.....	41	31		
			Total equivalent condensation	88	92	95	.....	130	124		
			Loss pressure friction	1.7	1.4	1.2	10.3	8.7	7.3	41.8	34.
			" " condensation	1.4	1.5	1.8	0.2	0.2	0.2	0.1	0.
3"	56.1	764,700	" " total	3.1	2.9	3.0	10.5	8.9	7.5	41.9	34.
			Condensation per hour	125	132	138	125	132	138	125	13
			" equivalent friction per hour	.....	.....	.....	7	5	4	67	4
			" " " total per hour	125	132	138	132	137	142	192	17
			Loss pressure friction	.....	.....	.....	2.3	2.0	1.7	9.3	7.
			" " condensation	.....	.....	.....	0.3	0.3	0.4	0.1	0.
4"	57.8	3,421,200	" " total	.....	.....	.....	2.6	2.3	2.1	9.4	7.
			Condensation per hour	.....	.....	.....	162	171	179	162	17
			" equivalent friction per hour	.....	.....	.....	1	1	1	12	
			" " " total per hour	.....	.....	.....	163	172	180	174	18
			Loss pressure friction	.....	.....	.....	0.7	0.6	0.5	3.0	2.
			" " condensation	.....	.....	.....	0.7	0.8	1.1	0.2	0.
5"	58.4	10,659,400	" " total	.....	.....	.....	1.4	1.4	1.6	3.2	2.
			Condensation per hour	.....	.....	.....	224	236	247	224	23
			" equivalent friction per hour	.....	.....	.....	.....	.....	.....	4	
			" " " total	.....	.....	.....	224	236	247	228	23
			Loss pressure friction	.....	.....	.....	.....	.....	.....	1.2	1.
			" " condensation	.....	.....	.....	.....	.....	.....	0.2	0.
6"	59.5	27,527,000	" " total	.....	.....	.....	.....	.....	.....	1.4	1.
			Condensation per hour	.....	.....	.....	.....	.....	.....	244	25
			" equivalent friction	.....	.....	.....	.....	.....	.....	1	
			" " " total	.....	.....	.....	.....	.....	.....	245	25
			Loss pressure friction	.....	.....	.....	.....	.....	.....	0.3	0.
			" " condensation	.....	.....	.....	.....	.....	.....	0.2	0.
8"	60.7	130,717,000	" " total	.....	.....	.....	.....	.....	.....	0.5	0.
			Condensation per hour	.....	.....	.....	.....	.....	.....	330	34
			" equivalent friction	.....	.....	.....	.....	.....	.....	.....	
			" " " total	.....	.....	.....	.....	.....	.....	330	34
			Loss pressure friction	.....	.....	.....	.....	.....	.....	.....	
			" " condensation	.....	.....	.....	.....	.....	.....	.....	
10"	61.8	381,960,000	" " total	.....	.....	.....	.....	.....	.....	.....	
			Condensation per hour	.....	.....	.....	.....	.....	.....	.....	
			" equivalent friction	.....	.....	.....	.....	.....	.....	.....	
			" " " total	.....	.....	.....	.....	.....	.....	.....	
			Loss pressure friction	.....	.....	.....	.....	.....	.....	.....	
			" " condensation	.....	.....	.....	.....	.....	.....	.....	
12"	62.1	959,496,000	" " total	.....	.....	.....	.....	.....	.....	.....	
			Condensation per hour	.....	.....	.....	.....	.....	.....	.....	
			" equivalent friction	.....	.....	.....	.....	.....	.....	.....	
			" " " total	.....	.....	.....	.....	.....	.....	.....	
			Loss pressure friction	.....	.....	.....	.....	.....	.....	.....	
			" " condensation	.....	.....	.....	.....	.....	.....	.....	
14"	62.3	2,087,305,000	" " total	.....	.....	.....	.....	.....	.....	.....	
			Condensation per hour	.....	.....	.....	.....	.....	.....	.....	
			" equivalent friction	.....	.....	.....	.....	.....	.....	.....	
			" " " total	.....	.....	.....	.....	.....	.....	.....	

ON LOSSES IN STEAM PIPE PER 1,000 FEET.

bs. = 0.218.
bs. = 0.261.
bs. = 0.304.

Condensation taken at 0.0005 lb. per hour per square foot covered external surface per degree difference in temperature.
Calculations to 60° external temperature.
Equivalent condensation due to friction to water at temperature of steam.
Loss pressure due to condensation by Marriotte's Law.

5,000 Lbs.			10,000 Lbs.			15,000 Lbs.			20,000 Lbs.			30,000 Lbs.			40,000 Lbs.		
319.2 100	275.0 120		796.6 80	638.3 100	550.0 120	1,149.9 80	957.5 100	825.0 120	1,533 80	1,277 100	1,100 120	2,300 80	1,915 100	1,650 120	3,066 80	2,553 100	2,200 120
34.7 0.1	30.1 0.1																
34.8	30.2																
132 45	138 34																
177	172																
7.8 0.1	6.7 0.1		37.5	31.1	26.9	.....	70.0	60.5									
7.9	6.8		37.5	31.1	26.9	.....	70.0	60.5									
171 9	179 7		162 116	171 78	179 57	....	171 228	179 175									
180	186		278	249	236	.....	399	354									
2.5 0.2	2.2 0.2		12.0 0.1	10.0 0.1	8.6 0.1	27.0	22.5	19.4	48.1	39.9	34.5	.....	89.9	77.6			
2.7	2.4		12.1	10.1	8.7	27.0	22.5	19.4	48.1	39.9	34.5	.....	89.9	77.6			
236 3	247 3		224 33	236 23	247 17	224 117	236 81	247 60	224 370	236 206	247 152	....	236 588	247 422			
239	250		257	259	264	341	317	307	594	442	399	.....	824	669			
1.0 0.2	0.8 0.2		4.7 0.1	3.9 0.1	3.3 0.1	10.5	8.7	7.5	18.7	15.5	13.4	41.8	34.8	30.0	74.6	61.8	53 5
1.2	1.0		4.8	4.0	3.4	10.5	8.7	7.5	18.7	15.5	13.4	41.8	34.8	30.0	74.6	61.8	53.5
257 1	268 1		244 12	257 9	268 7	244 43	257 30	268 23	244 103	257 74	268 55	244 403	257 265	268 193	244 1,272	257 546	268 439
258	269		256	266	275	287	287	281	347	331	323	647	522	461	1.516	803	707
0.2 0.3	0.2 0.3		1.0 0.1	0.9 0.1	0.8 0.1	2.4 0.1	2.0 0.1	1.7 0.1	4.2	3.5	3.0	9.5	7.9	6.9	17.0	14.1	12.2
0.5	0.5		1.1	1.0	0.9	2.5	2.1	1.8	4.2	3.5	3.0	9.5	7.9	6.9	17.0	14.1	12.2
347	366		330 2	347 2	366 2	330 8	347 7	366 5	330 20	347 15	366 12	330 77	347 54	366 41	330 188	347 134	366 99
347	360		332	349	368	338	354	371	350	362	378	407	401	407	518	481	465
....	....		0.3 0.2	0.3 0.2	0.2 0.2	0.8 0.1	0.6 0.1	0.5 0.1	1.3 0.1	1.1 0.1	1.0 0.1	3.0	2.5	2.2	5.4	4.5	3.9
.....	.....		0.5	0.5	0.4	0.9	0.7	0.6	1.4	1.2	1.1	3.0	2 5	2.2	5.4	4.5	3.9
....	....		415 1	437 1	467 1	415 2	437 2	467 2	415 6	437 5	467 4	415 21	437 17	467 14	415 53	437 40	467 31
.....	.....		416	438	468	417	439	469	421	442	471	436	454	481	468	477	498
....	....		.....	.....	.....	0.3 0.1	0.3 0.1	0.2 0.1	0.5 0.1	0.5 0.1	0.4 0.1	1.2	1.0	0.9	2.1	1.8	1.5
.....	.....		.....	.....	.....	0.4	0.4	0.3	0.6	0.6	0.5	1.2	1.0	0.9	2.1	1.8	1.5
....	....		.....	.....	.....	527 1	555 1	581 1	527 2	555 2	581 2	527 8	555 7	581 5	527 19	555 16	581 13
.....	.....		.....	.....	.....	528	556	582	529	557	583	535	562	586	546	571	594
....	....		.....	.....	.....	.....	.....	.....	0.3 0.1	0.2 0.1	0.2 0.1	0.6 0.1	0.5 0.1	0.4 0.1	1.0	0.8	0.7
.....	.....		.....	.....	.....	.....	.....	.....	0.4	0.3	0.3	0.7	0.6	0.5	1.0	0.8	0.7
....	....		.....	.....	.....	.....	.....	.....	620 1	652 1	682 1	620 4	652 3	682 3	620 9	652 8	682 6
.....	.....		.....	.....	.....	.....	.....	.....	621	653	683	624	655	685	629	660	688

Plant.	Type of Boiler.	Evaporation from and at 212° per Lb. Coal.	Coal per Boiler H.-P. per Hour.	Coal per Day 10 Hours.	Coal per Day 24 Hours.	Coal per Year 275 Days, 10 Hours per Day.	Coal per Year 365 Days, 24 Hours per Day.	Cost per Boiler H.-P. per Year 275 Days, 10 Hours per Day.	Cost per Boiler H.-P. per Year 365 Days, 24 Hours per Day.	Cost of 1,000 Lbs. Steam per Hour, per Year 275 Days, 10 Hours per Day.	Cost of 1,000 Lbs. Steam per Hour, per Year 365 Days, 24 Hours per Day.
		Lbs.	Lbs.	Lbs.	Lbs.	Tons.	Tons.	$	$	$	$
Williamstown.	B. & W.	7.51	4.594	45.94	110.25	5.740	17.967	5.81	14.98	193.67	499.33
	Cylinder.	6.13	5.628	56.28	135.07	6.911	22.011	6.68	18.01	222.67	600.33
	Entire Plant Inc. Climax.	7.22	4.778	47.78	114.67	5.867	18.687	5.90	15.52	196.67	518.40
Bear Valley Shaft.	B. & W.	8.15	4.233	42.33	101.59	5.198	16.555	5.40	13.97	180.00	465.67
	Cylinder.	6.71	5.143	51.43	123.43	6.315	20.014	6.24	16.59	208.00	553.00
	Entire Plant	7.66	4.504	45.04	108.10	5.531	17.615	5.65	14.71	188.33	490.33
Bear Valley Slope.	Return Tubular.	7.70	4.481	44.81	107.54	5.503	17.525	5.63	14.64	187.67	488.00
Lykens.	B. & W.	7.48	4.612	46.12	110.69	5.664	18.038	5.75	15.03	191.67	501.00
	Cylinder.	6.33	5.450	54.50	130.80	6.693	21.315	6.52	17.49	217.33	583.00
	Entire Plant	6.48	5.324	53.24	126.27	6.538	20.822	6.40	17.12	213. 3	570.67
Averages.	B. & W. and Tubular	.....	.....	.....	.....	.....	.....	5.65	14.65	188.25	488.50
	Cylinder.	.....	.....	.....	.....	.....	.....	6.48	17.36	216.00	578.78

The coal consumption and cost are calculated for 10 hours per day, 275 days per year, as well as for the full year, as the former is about the average working time of these collieries, and a considerable proportion of the steam is used only for that time; in the cost for the working year, however, the full charge for interest, etc., is included, as sufficient plant must be maintained to supply the maximum demand for steam.

The last column in tables Nos. 5 and 6 gives the cost per annum for steam in excess of 40 pounds per I. H.-P. per hour for each engine and pump, it being considered that 40 pounds should be readily reached even under the disadvantages of colliery practice. Most of the engines' valves could be changed to reach this figure, and while unattainable with the present direct-acting pumps, very many of these could be readily compounded by adding a high-pressure cylinder, the low mean effective pressure in most making this entirely feasible. From these figures of cost the engines and pumps worth improving can be readily chosen.

Steam Lines.—The detail dimensions of steam lines are given in tables Nos. 9 and 10, with the calculated condensation losses from the same; they are so interconnected in the mines that not even a fairly approximate estimate of the quantity of steam carried by each line can be made.

Condition of Steam Lines.—The steam lines as far as practi-

cable were all examined, and with very few exceptions found well covered with non-conducting material.

LYKENS VALLEY LINES: *Receivers; Traps.*—The lines of the Lykens Valley Coal Co. (shown on maps A, Fig. 6, and B, Fig. 7) were found in uniformly good condition, but five leaks being noted, three of these in expansion joints; these lines all run to receivers at each level, which act as separators to remove the water of condensation and which are all provided with home-made traps.

The condition of the steam on the various levels was found as follows by calorimeter:

No. 1 Level,	2.03	per cent.	moisture,	near	No. 3 Pump.
No. 2 "	0.8	"	"	"	Receiver.
No. 3 "	0.5	"	"	"	"
No. 4 "	2.0	"	"	"	No. 2 Pump.

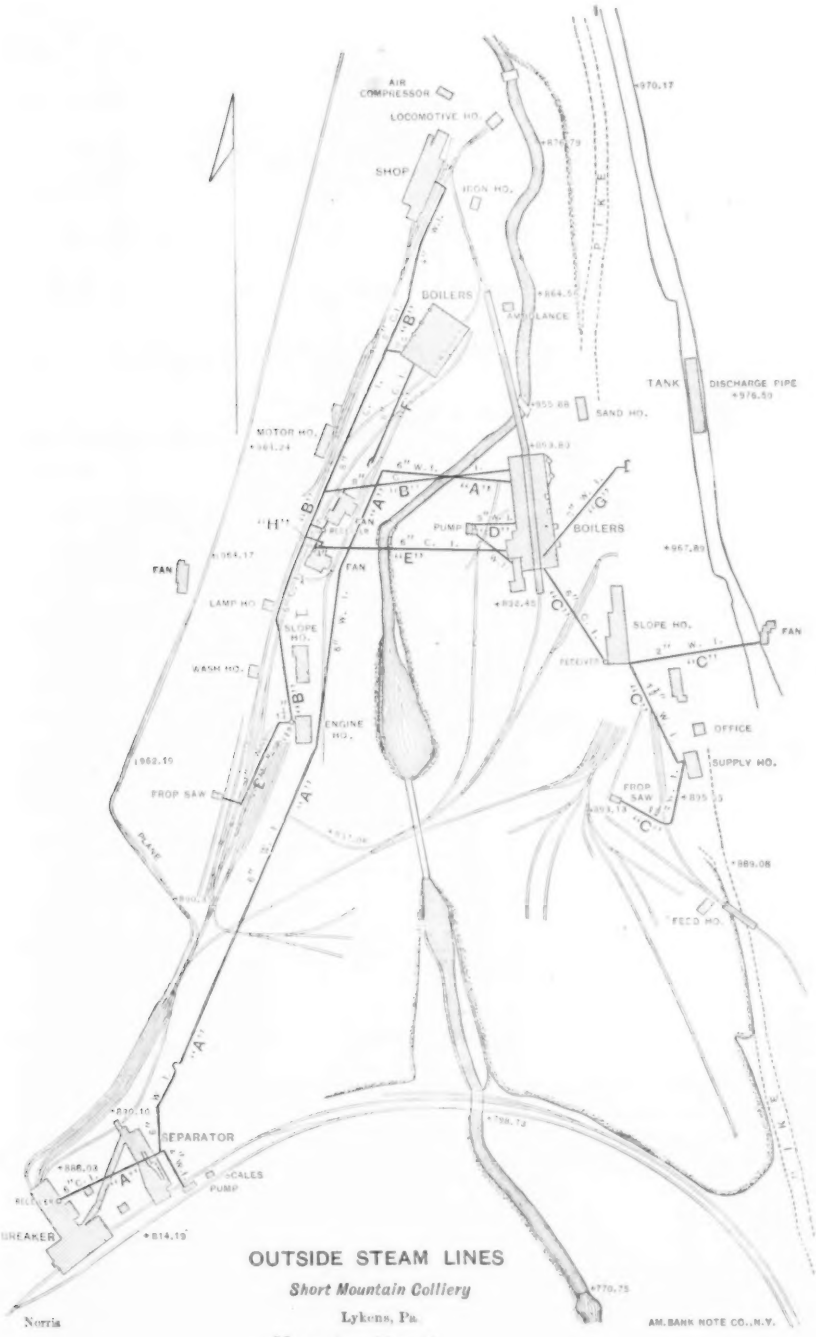
It is manifestly advantageous to remove the water of condensation from the steam lines as promptly as possible, as owing to its greater conductivity it must increase the radiation from the pipes, and it certainly is not conducive to keeping tight points.

The 8-inch high-pressure line "G" was out for repairs and not examined.

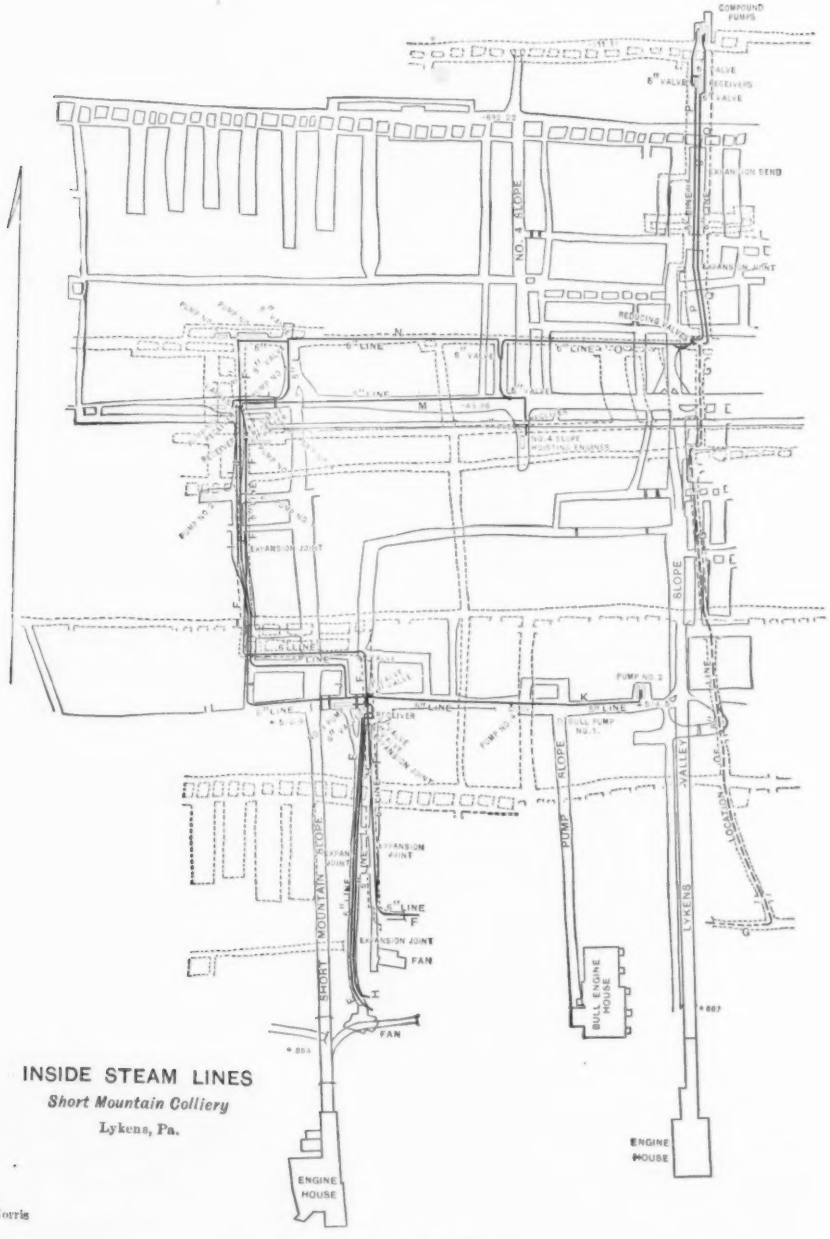
WILLIAMSTOWN LINES.—The steam lines in the Williamstown Colliery (Maps C, Fig. 8, and D, Fig. 9) were found well covered, but in many cases the inside lines were leaking badly, and making the pipe ways so hot that examination was impracticable, the temperatures along lines examined reaching over 125 degrees Fahr., and being much higher along other lines a full examination of which was not attempted. Most of the leaks were through the threads of flanges; all of these threaded flanges should be removed as rapidly as possible, and replaced by flanges secured by expanding the pipe into them. Several lines with flanges secured in this way were examined and not a single leak past the flanges discovered.

Lines in Bear Valley (Map E, Fig. 10) are all in good condition, except those down Summit Slope, which need some attention; the line down Bear Valley Shaft is more or less wet, causing considerable extra condensation.

The Williamstown Lines except in a very few instances are not provided with receivers or traps, the water of condensation being blown off from tees and partly open valves into the sumps, resulting in a great loss of steam and considerable undesirable addition to the temperature.


AIR COMPRESSOR
LOCOMOTIVE HO.
SHOP
IRON HO.
BOILERS
AMBULANCE
TANK
DISCHARGE PIPE
SAND HO.
MOTOR HO.
"H"
FAN
PUMP
"D"
"E"
"A"
"B"
"C"
"G"
BOILERS
LAMP HO.
SLOPE HO.
WASH HO.
ENGINE HO.
OFFICE
SUPPLY HO.
PROP SAW
FEED HO.
SEPARATOR
SCALES
PUMP
BREAKER
Norris
OUTSIDE STEAM LINES
Short Mountain Colliery
Lykens, Pa.
AM. BANK NOTE CO., N.Y.

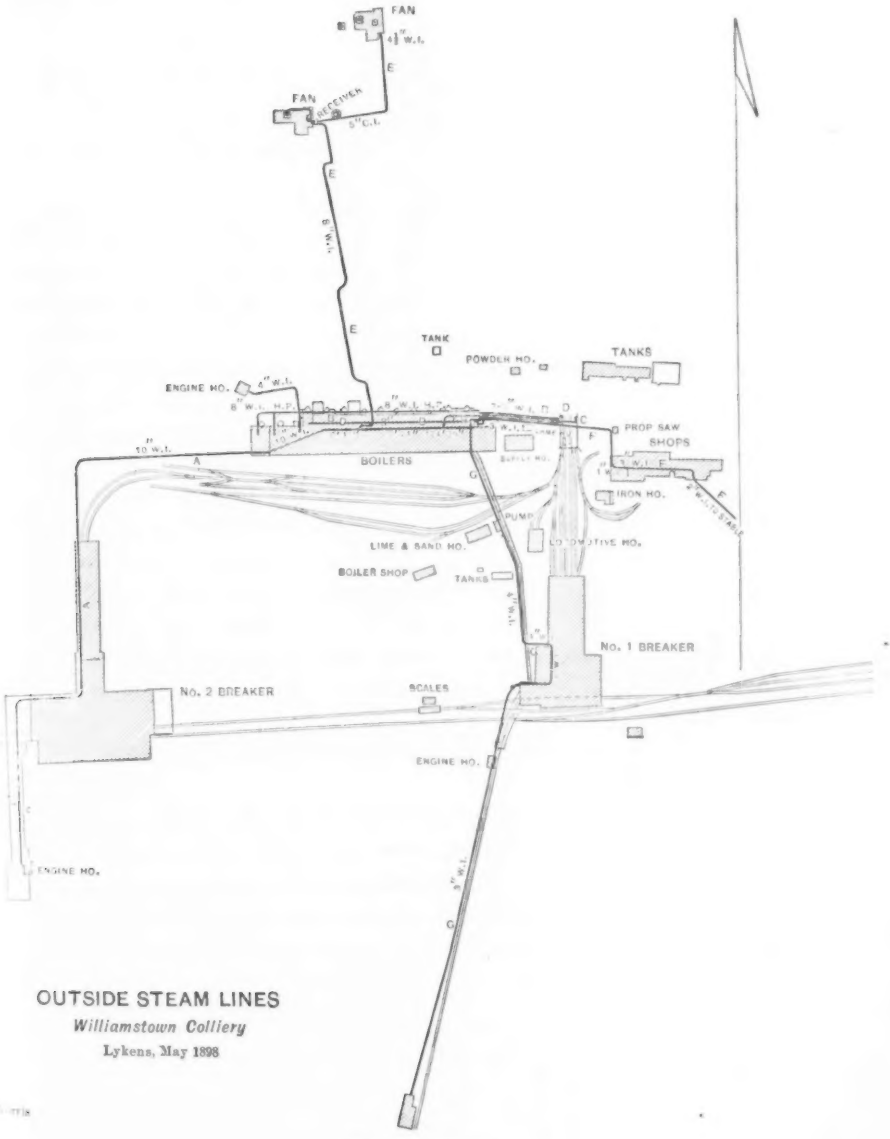
MAP A.—FIG. 6.



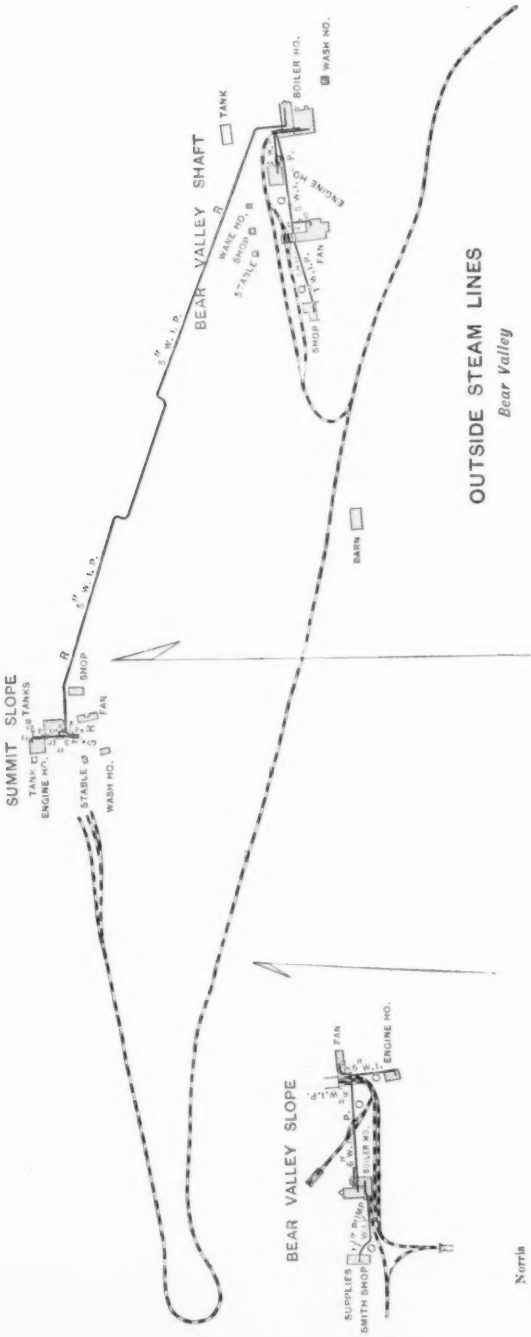
INSIDE STEAM LINES
Short Mountain Colliery
Lykens, Pa.

Norris

MAP B.—FIG. 7.


FAN
FAN
TANK
POWDER HO.
TANKS
ENGINE HO.
PROP SAW
SHOPS
BOILERS
IRON HO.
PUMP
LIME & SAND HO.
BOILER SHOP
TANKS
No. 1 BREAKER
No. 2 BREAKER
SCALES
ENGINE HO.
ENGINE HO.
OUTSIDE STEAM LINES
Williamstown Colliery
Lykens, May 1898

MAP C.—FIG. 8


OUTSIDE STEAM LINES
Bear Valley
BEAR VALLEY SHAFT
SUMMIT SLOPE
BEAR VALLEY SLOPE
TANK
BOILER HO.
WASH HO.
ENGINE HO.
SHOP
STABLE
FAN
BARN
SMITH SHOP
SUPPLIES
Norris

MAP E.—FIG. 10.

Condition of Steam.—The condition of steam at the various levels was found as follows:

No. 2 Pocket.....	$2\frac{3}{10}$	per cent.	moisture,	low	pressure	line.
No. 3 Slope Bottom.....	$\frac{8}{10}$	"	"	high	"	"
No. 4 " "	$3\frac{8}{10}$	"	"	"	"	"
Summit Slope Bottom....	$19\frac{2}{10}$	"	"			
Bear Valley Slope Bottom,	$3\frac{5}{10}$	"	"			

The pressure in the 8-inch high-pressure line at the foot of No. 4 Slope had fallen to 64 pounds; this line carries all the high-pressure steam from the boilers, amounting to over 40,000 pounds per hour, and is overtaked. A portion of this steam could with advantage be admitted into the low-pressure system at the boiler house, through a reducing valve, as is done now inside the mines, the pumps on this line not using all its steam. In the boiler test made with separator coal this is strongly evinced, the Climax waste heat boilers more than doubling their output and that of the Babcock & Wilcox being correspondingly reduced, while maintaining full pressure, showing that the pipe was carrying all it could take.

Friction and Condensation Losses.—The table No. 12 gives the losses per 1,000 feet of various-sized pipes due to friction and condensation, and will, we think, be of value in proportioning future pipes; in using it, it should be noted that the total equivalent condensation is the measure of the loss in heat units, but the economy of use of steam in engines increases with its pressure.

Condensation.—No very satisfactory figures were found in the text-books for the absolute condensation losses in pipes, most of the figures given being comparative with hair felt. The 0.012 pounds per 24 hours per square foot pipe degree Fahr. difference in temperature of steam and external air used in the calculations is based on the following:

Test by	Location.	Sq. Ft. Surf.	Lbs. Water in 24 Hours.	Lbs. Water per Sq. Ft. 24 Hours.	Different Temperature	Lbs. Water per Deg. per 24 Hours.	Covering.
Bedle & Bauer.	Line G. L. V. C. Co.	4,130	11,315	2.74	262	.0104	Asbestos.
Norris.....	I & J, L. V. C. Co..	3,892	9,360	2 40	234	0103	"
Brill.....	A. S. M. E.... ...	....	....	....	308	.0105	Magnesia sect'l.
Norton	A. S. M. E.....	....	.....	....	315	.0125	Magnesia.

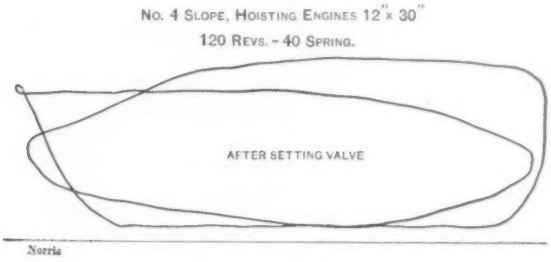
The last test from paper, by C. L. Norton (*Transactions* American Society Mechanical Engineers, June, 1898), was made with great care, and was given weight in deciding on .012 as a proper figure to be used in the calculations. In the course of Mr. Norton's experiments he found that a pipe boxed in with charcoal 1 inch minimum thickness was 20 per cent. better insulated than magnesia covering, corroborating the statements by Mr. Reinhart (general inside foreman Lykens Valley Coal Company) of his experience in using flue dust to insulate pipes. If this were packed in metal instead of wooden boxes, to avoid fire risk, it would seem to be an excellent non-conducting covering, and well worthy of a careful trial.

Engines and Pumps.—The sizes and performance of pumps and engines of the Lykens Valley Coal Company and the Summit Branch Coal Company are tabulated in tables Nos. 5 and 6 respectively. All of these were carded by either Mr. McKay or the writer, and copies of average indicator cards are given herewith. The total steam per horse-power per hour includes an allowance for cylinder condensation, based on data in Kent's "Pocket-book." The maximum speed given is the highest at which each has been continuously run not necessarily its absolute maximum.

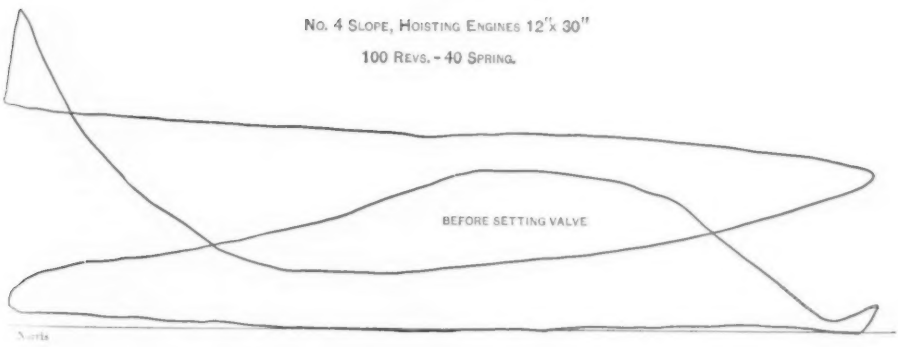
Fan Engines.—In examining these it is evident that all the fan engines are much too large, and are cutting off at nearly full stroke with a very low mean effective pressure, and using enormous quantities of steam; especially is this the case with the four 25-foot fans, with 24 × 36 inch engines, using from 75 to 86.5 pounds steam per horse-power per hour (cards 1 and 100). By supplying these engines with new valves, with more lap, or riveting on strips to increase the lap, to cut off at about ⅜ stroke, the steam consumption could probably be reduced to at least 40 pounds per horse-power, reducing the steam consumption for the seven fans, with slide-valve engines, from 16,600 pounds to at least 8,600 pounds per hour. The Bear Valley Shaft fan (card No. 49), with Corliss engines, shows 24.3 pounds indicated and 39.4 pounds total steam per horse-power per hour. The fan, however, is running but 52 revolutions and indicating 40.2 horse-power, while its designed speed is 85 revolutions, which would require about 177 horse-power, and the engine designed for the latter is cutting off at less than 5 per cent., far below its most economical point.

Summit Branch Coal Company. Card numbers as in Table No. 6.

No. 22.



No. 22.



No. 24.

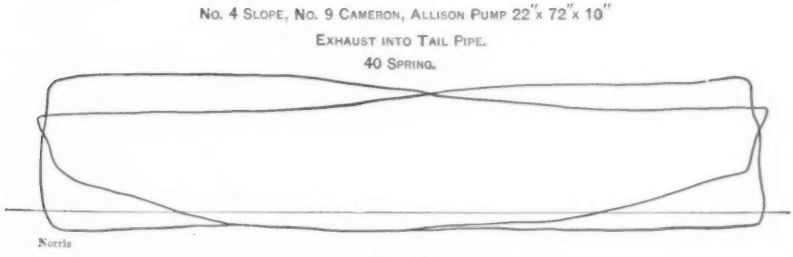
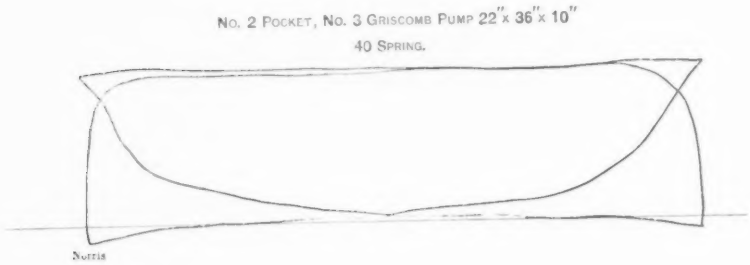


Fig. 11.

Summit Branch Coal Company. Card numbers as in Table No. 6.

Nos. 13 and 14.



No. 19.

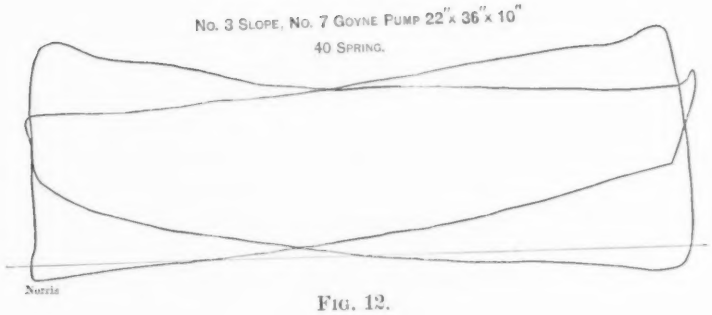


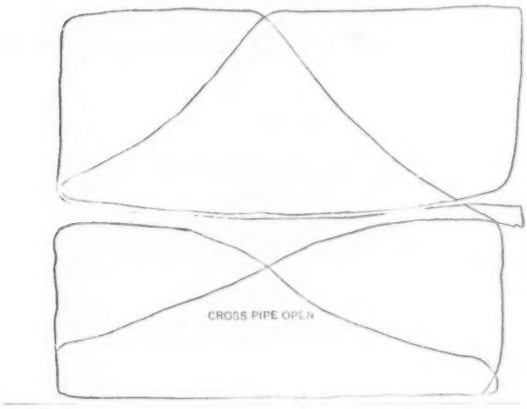
Fig. 12.

Summit Branch Coal Company. Card numbers as in Table No. 6

Nos. 47 and 48.

Greenfield Riedler Pump 13″& 20½″ x 16″ x 7 5/16″

96 Revs. -50 Spring.

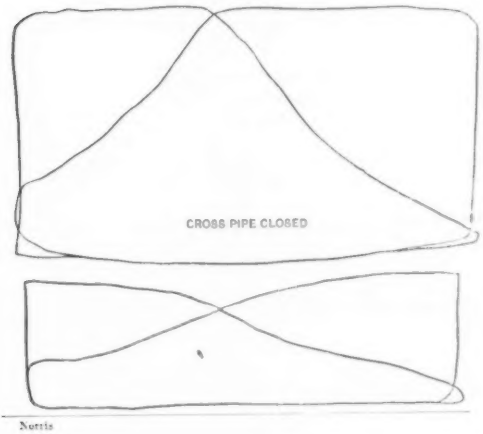


Norris

Nos. 47 and 48.

Greenfield Riedler Pump 13″& 20½″ x 16″ x 7 5/16″

85 Revs. - 50 Spring.

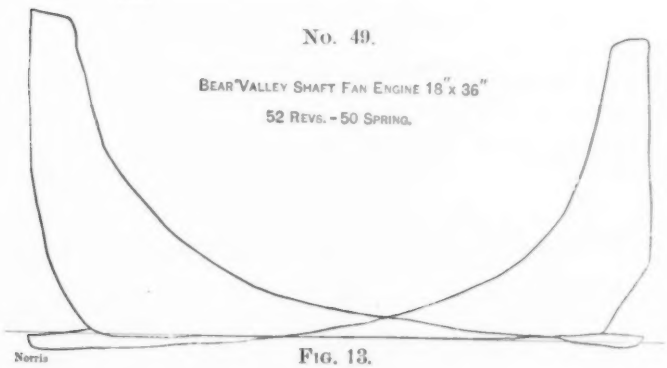


Norris

No. 49.

Bear Valley Shaft Fan Engine 18″ x 36″

52 Revs. - 50 Spring.

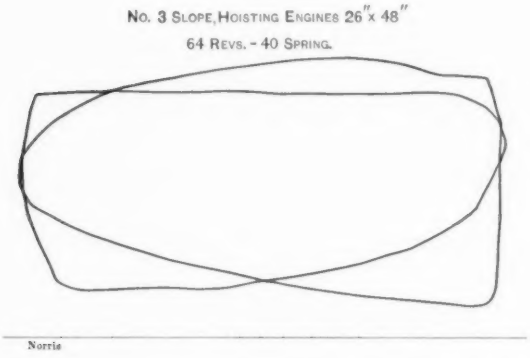


Norris

Fig. 13.

Summit Branch Coal Company. Card numbers as in Table No. 6.

No. 11.



No. 12.

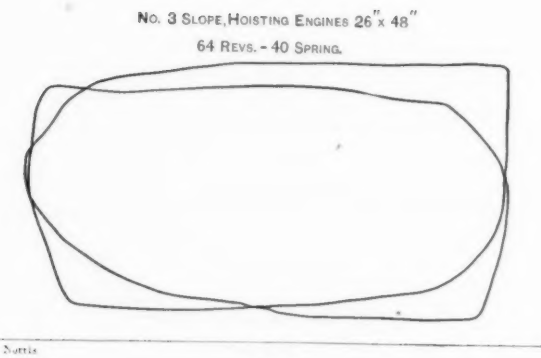
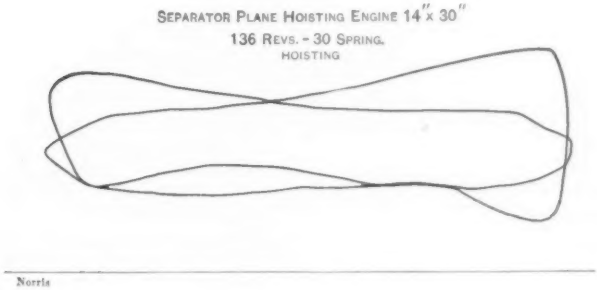


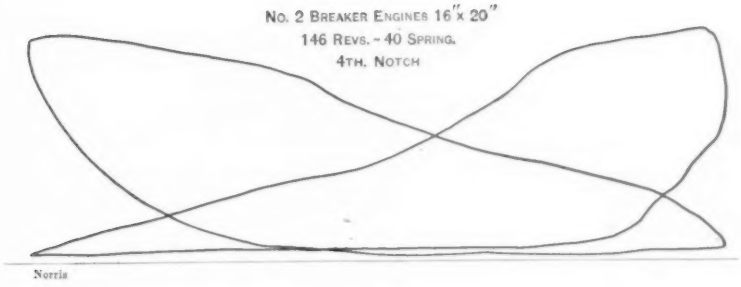
Fig. 14.

Summit Branch Coal Company. Card numbers as in Table No. 6

No. 42.



No. 45.



No. 45.

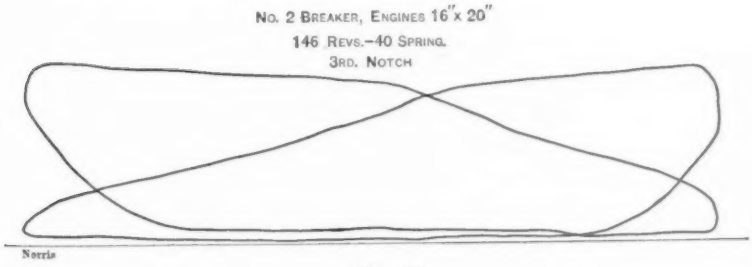
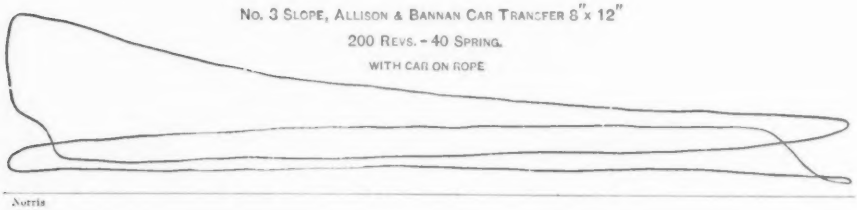


FIG. 15.

Summit Branch Coal Company. Card numbers as in Table No. 6.

No. 29.



No. 35.

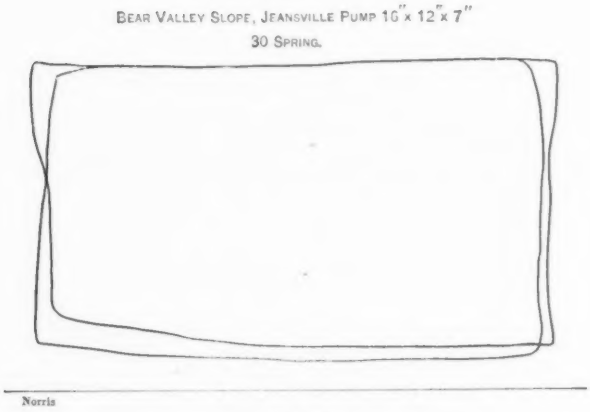
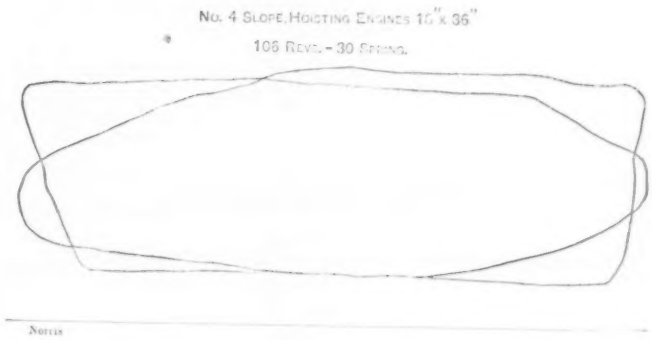


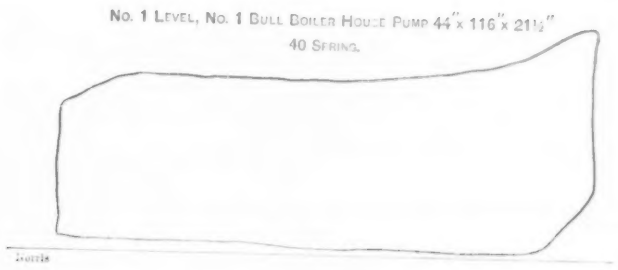
Fig. 16.

Lykens Valley Coal Company's Engines and Pumps. Card numbers correspond with those in Table No. 5.

No. 130.



No. 131.



No. 131.

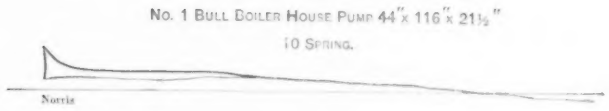


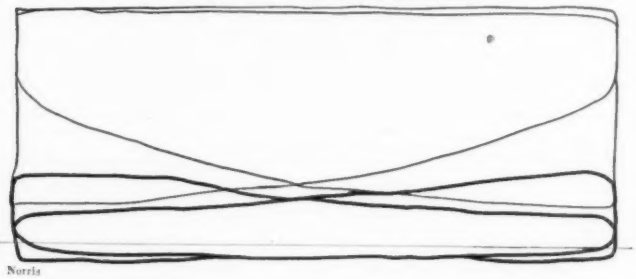
Fig. 17.

Lykens Valley Coal Company's Engines and Pumps. Card numbers correspond with those in Table No. 5.

Nos. 115–118.

No. 4 Level, No. 2 Jeansville Pump 25½" & 38" x 36" x 10"

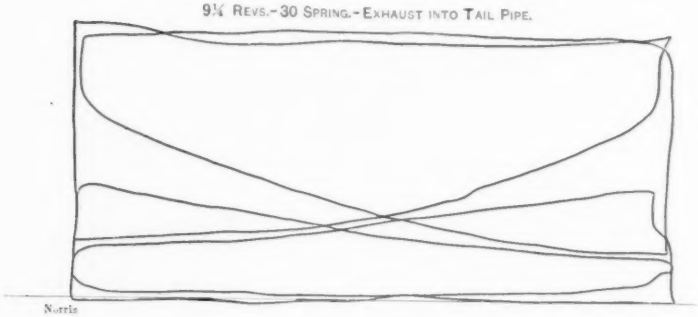
9½ Revs.–30 Spring.–Syphon Condenser.



Nos. 119 and 120.

No. 4 Level, No. 1 Jeansville Pump 24" & 38" x 36" x 10"

9¼ Revs.–30 Spring.–Exhaust into Tail Pipe.



No. 128.

No. 3 Level, No. 4 Allison Pump 30" x 69¾" x 14½"

30 Spring.

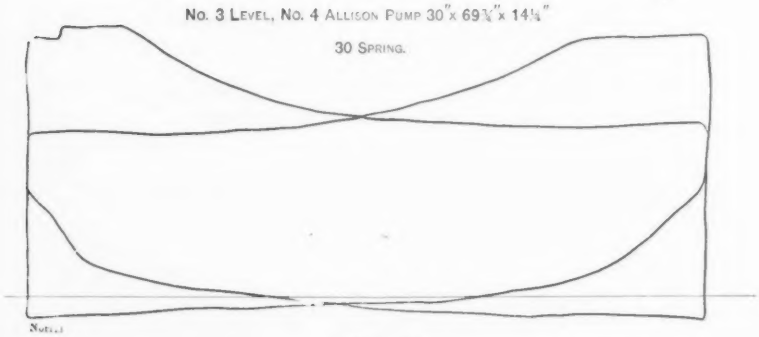
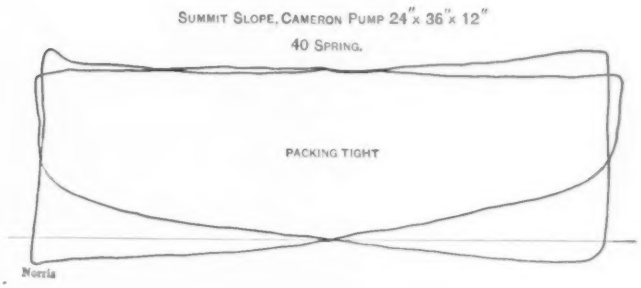


Fig. 18.

Summit Branch Coal Company. Card numbers as in Table No. 6.

No. 52.



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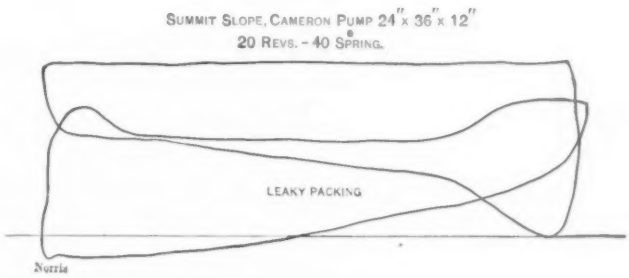


No. 59.

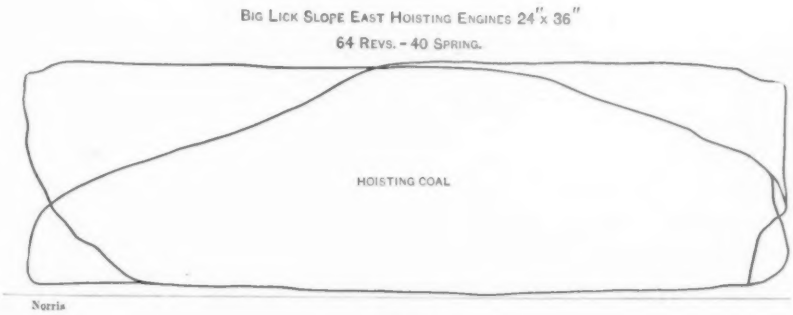
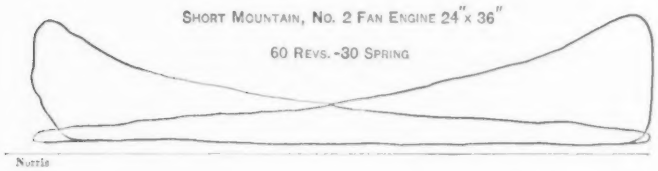


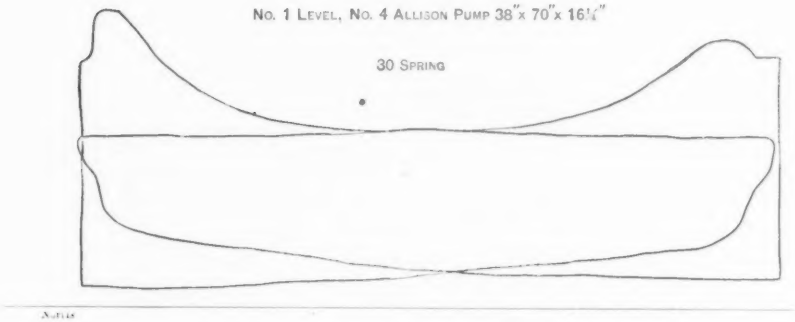
FIG. 19.

Lykens Valley Coal Company's Engines and Pumps. Card numbers correspond with those in Table No. 5.

No. 100.



No. 110.



Nos. 111 and 112.

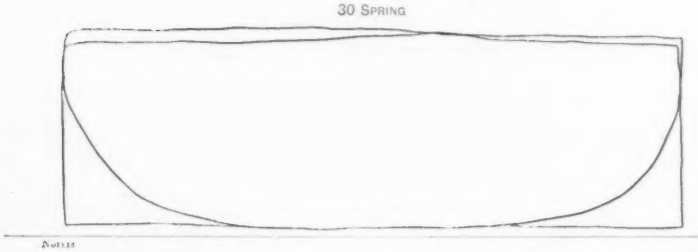
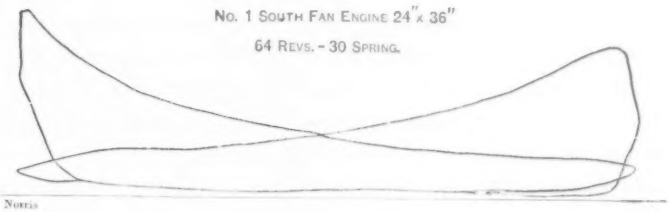


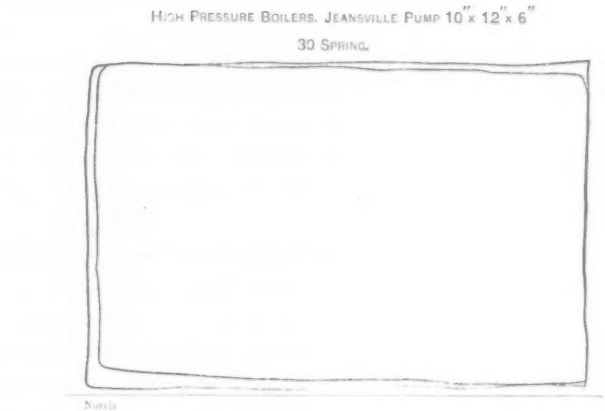
Fig. 20.

Summit Branch Coal Company. Card numbers as in Table No. 6.

No. 1.



Nos. 3 and 4.



Nos. 6 and 7.

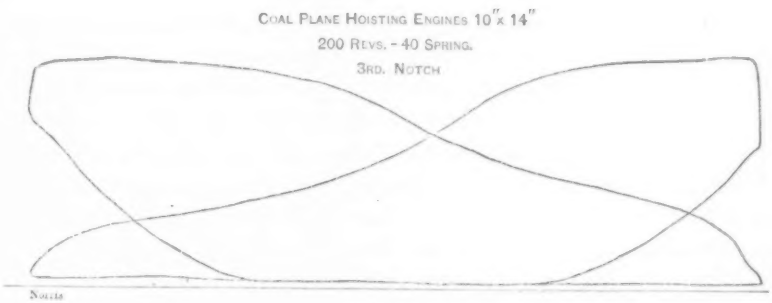


Fig. 21.

The No. 1 and No. 2 25-foot fans of both the Lykens Valley Coal Company and Summit Branch Coal Company are so arranged that either may do the work, the other standing idle, and it appears probable that considerable economy would result from running them in that manner, if inside conditions will permit.

Hoisting Engines.—Except the new engines at No. 2 Breaker Coal and Dirt Planes, Bear Valley Shaft and Big Lick Slope (cards Nos. 6 and 7 and 59), showing 32.7, 38.5, 39.6, and 39.1 pounds total steam per horse-power per hour, all the engines of both companies showed very poor steam economy; the valves of No. 4 Slope, Summit Branch Coal Company, engines (card No 22) were reset, improving them somewhat. All of these engines are of old pattern, with very small port areas; they could probably be somewhat improved by new valves, but not very materially.

Attention is especially called to the Short Mountain Slope engine, a single engine with fly-wheel and geared drum; this is in very bad condition—cylinder worn, bed plates cracked and foundation loosened—and if this slope is expected to have any considerable life it would certainly be advisable to get a new pair of modern engines for it.

The hoisting engine at the Williamstown Separator Plane is also worn out, the valve leaking very badly (see card No. 42, Table No. 8). The valve seat probably needs refacing, and with a new valve the engine would probably continue to do the little work required of it for a considerable time, with more nearly reasonable economy.

The Bear Valley Slope hoisting engines are now being replaced.

Miscellaneous Outside Engines.—All the outside stationary engines except those in No. 2 Breaker, Summit Branch (cards No. 45), and Lykens Separator, show excessive steam consumption, which could be much reduced by increasing the lap of the valves. Especial attention is called to the car transfer engine at the head of No. 3 Slope, Williamstown, which consumes steam at the rate of 201 pounds per horse-power per hour, and badly needs a new valve. (Card No. 29. The card given is with cars on the rope; the engine runs empty most of the time.) The No. 2 Breaker engines show better ecenomy cut off at the fourth notch of the reverse lever, where they were designed to

run, than at the third notch, where it was necessary to run them, on account of heating, the total steam consumption being 35.3 and 38.5 pounds per horse-power per hour respectively (card Nos. 45).

Valves.—In considering the advisability of making new valves for many of these engines, it must be remembered that in almost all cases the eccentrics are keyed fast and will require resetting, and that the valve rods have no adjustments outside the steam chests, and in the older small engines no adjustments of any sort, except by cutting and rewelding the rods.

Outside Pumps.—The outside pumps for boiler feed are all doing as well as could be expected of pumps of their type (card Nos. 3 and 4).

Fresh-water Pumps.—At the Greenfield plant the Riedler pump shows the best steam economy found, viz., 27.7 pounds total steam per horse-power per hour; the pump was, however, being run with the starting pipe partly open, feeding live steam to the low-pressure cylinders, to enable the required speed to be maintained. This could be done with better economy by raising the boiler pressure. The cards Nos. 47 and 48 show the effect of opening this cross pipe; with it closed, and at lower speed, the steam consumption was 1.2 pounds lower on a much less power. The Cameron pump at this station, used for emergencies, is in bad condition, the crib foundations very weak, and the pump pounding badly.

The two pulsometers at Lykens breaker were roughly tested by measuring by weir the quantity of water reaching them, and taking its temperatures before and after passing, and showed a consumption of over 700 pounds of steam per horse-power per hour. It seems as though some more economical means of raising this dirty and gritty water might be devised.

Mine Pumps.—The mine pumps use a very large proportion of the steam generated at these collieries—about 50 per cent. at Lykens Valley, 64 per cent. at Williamstown, 82 per cent. at Bear Valley Slope, and 81 per cent. at Bear Valley Shaft.

The mine pumps at Lykens are generally in good condition, though, except the two Jeansville compounds, at the foot of No. 4 Slope, of antiquated type. The Williamstown pumps are not as good, many of them having badly worn valve motions and plungers, and a number with bare steam cylinders.

Only a few of the pump cards have been engraved, but Nos.

13, 19, 24, 35, and 52 of the Summit Branch Coal Company, and 110, 111, and 128 of the Lykens Valley Coal Company, are typical of the various pumps in use.

Cards No. 131 are from the two ends of the single-acting Cornish "Bull" pump, the steam pressure raising the rods, which act on the plunger by weight on the return stroke.

Excepting the compounds, the Lykens Valley pumps average 55.5 pounds steam per horse-power per hour, and the Williamstown 61.2 pounds, the difference being mainly due to the use of syphon condensers on six of the twelve Lykens Valley pumps, while all the Williamstown exhaust into the tail pipes. All the Lykens Valley pumps are provided with counters, which are read daily, giving a complete record of the work done, while none of the Williamstown pumps are thus equipped, and the table of performance is necessarily based on counting at intervals, which was done during the boiler tests.

Compound Pumps.—The economy of compound pumps is very marked, the two compounds using respectively 37.7 (card Nos. 115–118) and 39.4 (card Nos. 119 and 120) pounds of steam per horse-power per hour, exhausting into syphon condenser and tail pipe, while the average of all the other mine pumps is about 58.5 pounds, so that by replacing all the pumps with compounds, a saving of about 34 per cent. of the steam should be attained, amounting per day of average work to 220,000 pounds for Lykens, 295,000 pounds for Williamstown, and 105,000 pounds for Bear Valley Shaft and Slope, or a total of 620,000 pounds per day, costing about $14.50 for Lykens Valley and $23.30 for Williamstown, or about $13,800 per year for the two companies, and compound pumps could readily be made to handle the water in two lifts instead of in four in each colliery.

Pump Valves.—The steam valves were generally found well set, and where they were not were left in that condition, with but few exceptions, as the cards herewith show. Attention is called to cards No. 52, Summit Branch, showing the effect on a slow-moving pump of a small leak in the steam packing.

Water Ends.—No cards were taken on the water ends of any of the pumps.

Speed.—The maximum speeds of the various pumps given on the sheets is the highest speed at which they are stated ever to have been run for any length of time; not necessarily the highest that they could be run in an emergency. The tabulated

steam consumption at maximum speed is in proportion to the increase of speed. In the absence of data it was believed that the increased resistance of the columns at higher speeds would be about balanced by the decrease in steam consumption per horse-power attending the increased pressure. The "test speed" given in Tables 5 and 6 is about the average for the year.

Pumping Capacity.—The plunger displacement of the various plants in gallons per minute is as follows, the actual capacity involving an unknown and variable allowance for slip:

LYKENS VALLEY COAL COMPANY.

No. 1 Level.

Pump No.	Diameter of Plunger.	Plunger Speed, Feet per Minute.		Gallons per Minute.	
	Inches.	Test.	Max.	Test.	Max.
1	$21\frac{1}{2}$	40.3	45.9	760	866
2 Dup.	$9\frac{3}{16}$	234.0	303.0	805	1,032
3	$11\frac{9}{16}$	.....	156.4	...	852
4	$16\frac{1}{4}$	70.0	93.3	754	1,005
Total No. 1 Level to surface.....				2,319	3,755

No. 2 Level.

1	$16\frac{1}{4}$	40.8	105.0	441	1,131
2	$12\frac{1}{4}$	....	107.6	...	658
3	$16\frac{1}{4}$	67.0	105.0	721	1,131
4	$12\frac{3}{8}$	107.8	138.0	701	897
Total No. 2 Level.....				1,873	3,817

No. 3 Level.

1	$12\frac{1}{8}$	....	115.4	...	692
2	$12\frac{3}{8}$	73.6	113.3	478	737
3	16	66.6	104.0	768	1,086
4	$14\frac{1}{4}$	98.8	139.5	819	1,156
Total No. 3 Level.....				2,065	3,671

No. 4 Level.

1	10 Dup.	.....	210.0	...	857
2	10 Dup.	111.0	210.0	453	857
Total No. 4 Level.....				453	1,714
Total pumped to surface, L. V. C. Co				2,319	3,755

SUMMIT BRANCH COAL COMPANY.

Plant.	Level.	Pump No.	Diameter of Plunger. Inches.	Plunger Speed, Feet per Minute. Test.	Max.	Gallons per Minute. Test.	Max.
WILLIAMSTOWN.	No. 2 Pocket	1	10	92.8	208.0	278	848
		2	12	133.0	175.0	781	440
		3	10 Dup.	292.8	450.0	1,194	1,836
	No. 3 Pocket	..	10	98.4	150.0	401	612
Total Upper Lift to surface						2,754	3,736
	No. 3 Slope	4	10	175.8	188.8	717	770
		5	9	158.8	201.5	525	666
		6	16	49.4	99.9	515	1,042
		7	10	106.8	150.0	436	612
Total Second Lift						2,193	3,090
	No. 4 Slope	8	8	186.3	222.3	486	580
		9	10	105.6	216.0	431	881
		10	16	43.8	84.0	457	877
		11	10 Dup.	128.8	200 0	526	816
Total Third Lift						1,900	3,154
B. V. SLOPE....	First Lift	..	9	102.7	146 7	339	485
		..	9	83.6	146.7	276	485
Total First Lift to surface						615	970
	Second Lift	..	7 Dup.	81.6	120.0	163	239
Total Bottom Lift						162	239
B. V. SHAFT...	Upper Lift	..	8 to am.	84.0	120.0	219	313
	Lower Lift	..	8	72.5	120.0	189	313
SUMMIT SLOPE..	Blake	..	12	54.0	120.0	317	705
	Niagara	..	9	102.7	146.7	339	485
	Cameron	..	12	108.0	180.0	634	1,058
Total Summit Slope to surface						1,290	2,248
Total pumped to surface, S. Br. C. Co.						4,878	7,267

Speed.—It will be noted that the Williamstown pumps are run at considerably higher speed than those of the Lykens Valley Coal Company, which probably helps to account for their worse condition.

Leakage and Slip.—The plunger displacement of the No. 3 Level L. V. pumps during the boiler tests was 2,065 gallons per minute, while the displacement for those at the No. 2 Level, which repumped all the No. 3 water and some additional, was but 1,873 gallons per minute; it would be advisable to carefully

examine the No. 3 Level pumps for slip, and the column pipe and the No. 2 Level sumps for leakage.

Cylinder Cocks.—The cylinder cocks in the mines were invariably found closed at Williamstown, and so slightly open at Lykens as to lose comparatively little steam through them, but at Williamstown, where the cocks were closed, separators on most pumps were found more or less open, blowing into the sumps. It seems probable that the use of automatic snifter valves on all engines and pumps would give a considerable saving of steam, if they were kept in order.

Hoisting Water.—The economy of hoisting water from the mines, instead of pumping, is very forcibly shown by the relative performance of the pumps and more modern hoisting engines, the pumps averaging nearly sixty pounds steam and the engines less than forty pounds per horse-power per hour, which difference is much increased in practice by the gain in finally disposing of water hoisted without loss by slip of pumps or leakage of columns. It would seem advisable to consider the question of hoisting the water from Bear Valley Shaft by cage tanks. Independent of the greater economy of hoisting over pumping water, the former has the great advantage of being free from the danger of drowning by heavy influxes of water.

Steam Used.—The steam generated and accounted for at each plant is shown in the schedule Table No. 11, the unaccounted for steam comprising all leakage except from safety valves, which was estimated, and all unaccounted for use: this difference amounts in the Lykens plant, where the lines were in good condition, with no apparent serious leakage, to $13\frac{4}{10}$ per cent., and it would seem that any difference exceeding about 15 per cent. must be due to preventable leaks. The $15\frac{1}{2}$ per cent. difference in the small plant at Big Lick is known to be due to leakage, which was estimated by comparing the escaping steam with the steam from the calorimeter at 2,200 pounds, nearly equalling the difference found, the 38 per cent. difference at Williamstown being unquestionably due to the large leaks and blow-offs in the mines.

Conclusion.—In conclusion, I want to express my indebtedness to Mr. T. M. Williams, superintendent Lykens Valley Coal Company and South Branch Coal Company, for the full facilities given me in making this investigation; to Mr. W. T. Reinhart, general inside foreman, and Mr. Frank Bauer, boiler inspector,

for their assistance and suggestions, Mr. Reinhart having accompanied me in the very trying inspection of the steam lines, and Mr. Bauer having been invaluable in conducting the boiler tests and having voluntarily taken night shift on the same; and, most of all, to Mr. Hood McKay, assistant engineer, to whose unremitting work and many and valuable suggestions are due the completeness of the data obtained. I am also indebted to Mr. Bedle, machinist Lykens Valley Coal Company, for assistance and data in regard to the steam lines, and to Mr. Radle, machinist Summit Branch Coal Company, for the same, Mr. Bedle's data being especially well arranged and valuable.

DISCUSSION.

Mr. Robert Allison.—I notice in Mr. Norris's paper that the Allison pump takes up considerable of the publication.

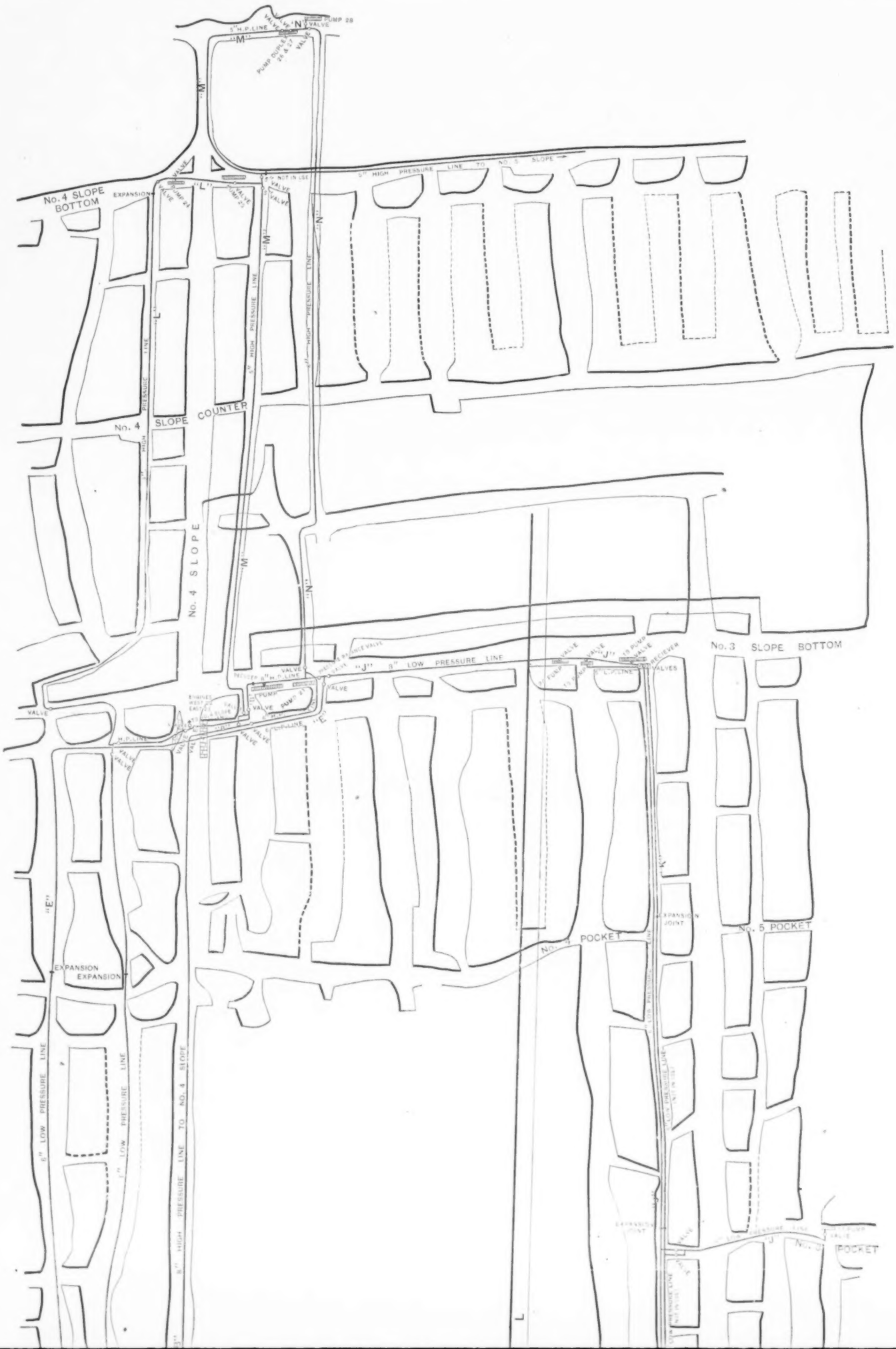
I would say that the first of the Allison pumps referred to were put in the mines about 1872, and most of those pumps, I suppose, have averaged about twenty years of almost constant use.

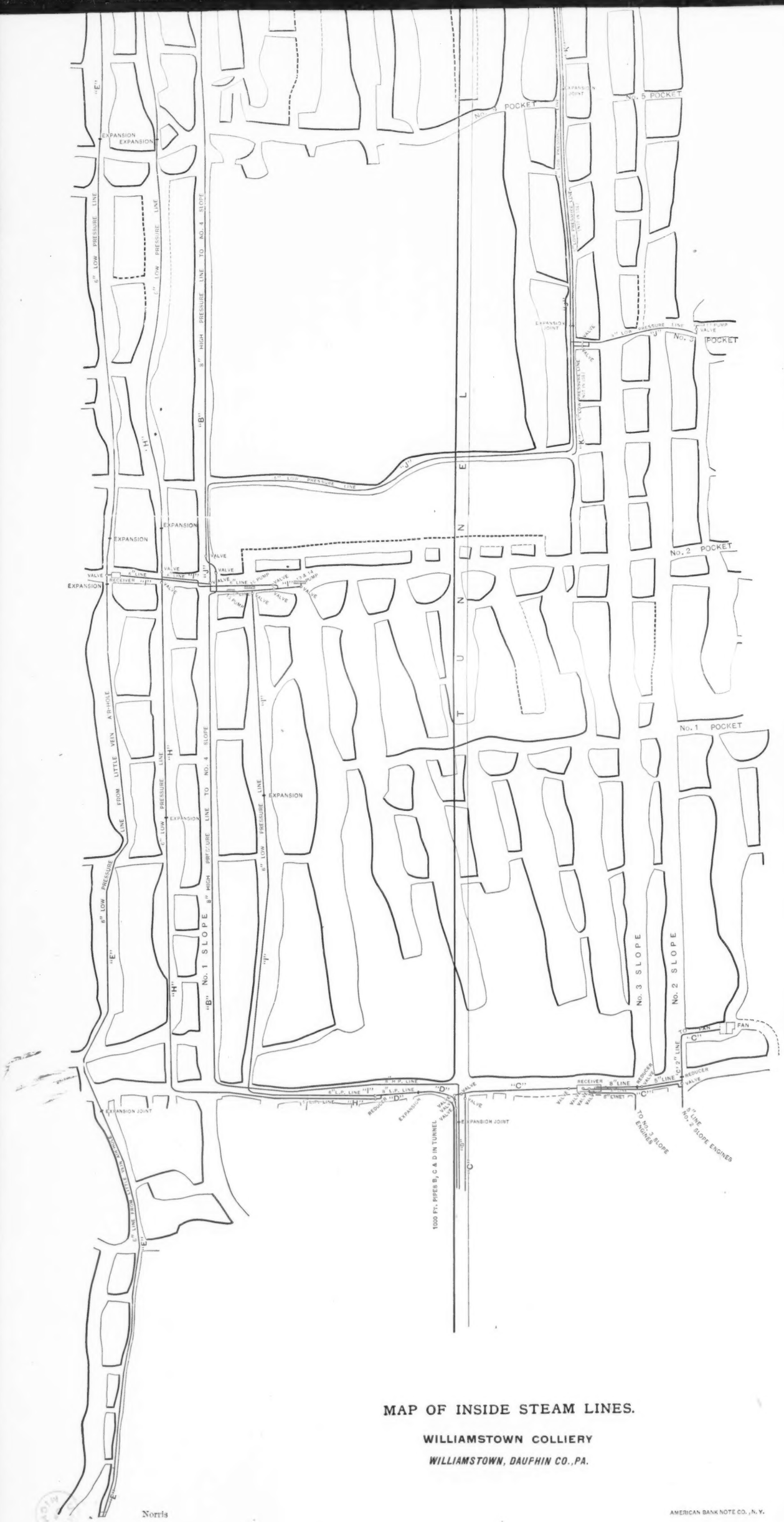
Mr. Norris, in his paper, speaks of some of the pumps being in pretty bad condition; those who know little about mine water cannot realize how mine pumps are used, and the difficulties they have to go through from acid in the water and carrying steam through long lines of steam pipe. I did not look over the tables of length of steam pipes, but suppose some of the pipe lines are 2,000 feet or more. In this connection I would like to quote from a circular prepared by permission of the *Colliery Engineer* from a letter addressed to them by one of the pump engineers in March, 1890. The pumps referred to are all of the regular Allison Cataract mining pump pattern. The article was written without the knowledge of the builders of the pumps.

"Short Mountain Colliery, belonging to the Lykens Valley Coal Company, is located in Wiconisco Township, the northeastern part of Dauphin County, in the north prong of the Dauphin fork, or the extreme western end of the anthracite coal measures. It is one of the many collieries in which, if you want to keep mining, you must have plenty of pumps and keep them in very good order to do your pumping.

"We have at this colliery fourteen (14) pumps of different styles, sizes, and capacities, viz.:

"One Bull, or Cornish pump, diameter of steam-cylinder 44 inches; diameter of water-cylinder 22 inches; stroke 10 feet; capacity 886 gallons; distance from steam-boilers 30 feet.


No. 4 SLOPE BOTTOM
No. 4 SLOPE COUNTER
No. 4 SLOPE
No. 3 SLOPE BOTTOM
No. 3 POCKET
No. 5 POCKET
No. 5 POCKET
EXPANSION JOINT
EXPANSION
EXPANSION
6" H.P. LINE
PUMP 28
PUMP DISCH. 25 & 27
6" HIGH PRESSURE LINE TO NO. 3 SLOPE
8" HIGH PRESSURE LINE
8" LOW PRESSURE LINE
6" LOW PRESSURE LINE
8" HIGH PRESSURE LINE TO NO. 4 SLOPE
"J" 8" LOW PRESSURE LINE
"M"
"N"
"L"
"E"
"K"



MAP OF INSIDE STEAM LINES.

WILLIAMSTOWN COLLIERY

WILLIAMSTOWN, DAUPHIN CO., PA.

Norris

AMERICAN BANK NOTE CO., N. Y.

"Three (3) Allison pumps, diameter of steam-cylinders 38 inches; diameter of water-cylinders 16¼ inches; stroke 6 feet; capacities 1,116 gallons each; distance from steam boilers, respectively 1,250, 1,680, and 2,300 feet.

"One Allison pump, diameter of steam-cylinder 30 inches; diameter of water-cylinder 14¼ inches; stroke 6 feet; capacity 864 gallons; distance from steam-boilers 2,300 feet.

"One Allison pump, diameter of steam-cylinder 30 inches; diameter of water-cylinder 12¼ inches; stroke 6 feet; capacity 630 gallons; distance from steam-boilers 1,700 feet.

"One Allison pump, diameter of steam-cylinder 27 inches; diameter of water-cylinder 12½ inches; stroke 6 feet; capacity 714 gallons; distance from steam-boilers 2,300 feet.

"One Allison pump, diameter of steam-cylinder 27 inches; diameter of water-cylinder 12¾ inches; stroke 6 feet; capacity 684 gallons; distance from steam-boilers 2,300 feet.

"One Allison pump, diameter of steam-cylinder 36 inches; diameter of water-cylinder 16¼ inches; stroke 6 feet; capacity 1,116 gallons; distance from steam-boiler 1,680 feet.

"One Allison pump, diameter of steam-cylinder 25 inches; diameter of water-cylinder 12¼ inches; stroke 6 feet; capacity 630 gallons; distance from steam-boilers 1,680 feet.

"One Griscom (Duplex) pump, diameter of steam-cylinder 22 inches; diameter of water-cylinders 10 inches; stroke 3 feet; capacity 1,225 gallons; distance from steam-boilers 1,420 feet.

"One Carter & Allen pump, diameter of steam-cylinder 26 inches; diameter of water-cylinder 12 inches; stroke 6 feet; capacity 560 gallons; distance from steam-boilers 1,000 feet.

"One Niagara (Hardick) pump, diameter of steam-cylinder 26 inches; diameter of water-cylinder 6 inches; stroke 3 feet; capacity 114 gallons; distance from steam-boilers 3,500 feet.

"One Cameron pump, diameter of steam-cylinder 18 inches; diameter of water-cylinder 8 inches; stroke 30¾ inches; capacity 280 gallons; distance from steam-boilers 3,100 feet.

"In the aggregate, when pumps are running at maximum speed, we are able to pump 11,275 gallons per minute, or about forty-seven (47) tons per minute.

"This is the amount that can be pumped over one section or pump-lift of about 315 feet, but we have five (5) sections or pump-lifts in the colliery equal to a vertical height of 1,576 feet. There are four (4) pumps on each section or pump-lift, but the two lower have only one pump on each section or lift. I will now give you the record of the four pumping over our upper section, commencing with the year 1886:

Year.	Gallons.	Cubic Feet.	Tons.
1886	704,738,176	94,209,791	2,944,055
1887	593,105,632	79,286,690	2,477,709
1888	531,939,234	71,109,933	2,222,186
1889	699,674,018	93,532,810	2,922,901
Totals	2,529,457,060	338,139,224	10,566,851

"Equal to a lake of water ten (10) feet deep, two hundred (200) feet wide, and thirty-two miles long; or, four years or 1,460 days, our average pumping was 1,732,505 gallons per day, equal to 248,963 cubic feet, or 7,780 tons, equal to 324 tons per hour, or nearly five and one-half (5½) tons per minute. During the same time we sent to market 852,831 tons of coal and pumped twelve and four-tenths ($12\frac{4}{10}$) tons of water to every ton of coal sent to market.

"This calculation is based on 2,000 pounds to a ton. You will understand me, this amount of water is pumped over one section or pump-lift of 315 feet, and the same water has to be handled three times and some of it five times. Also, while our average for four years has been 1,732,505 gallons per day, there have been times when we have pumped in twenty-four hours 4,500,000 gallons, equal to 18,798 tons, or about thirteen (13) tons per minute. I think there are very few collieries in the anthracite region that handle as much water as we do.

"I will conclude by giving you a complete record of one of the pumps of this colliery.

"The pump was purchased of Messrs. John Mullen & Son, Shamokin Iron Works in the fall of 1884. It was placed in position on our No. 3 level or pump station, and commenced running April 25, 1885. I did not keep a record of its work that year, but there was no expense incurred but the simple running wear-and-tear expenses. I kept a daily record for four years, beginning with the year 1886. It is an Allison pump, 38-inch steam-cylinder; 16¼-inch pole, and 6-foot stroke, working on a lift of 301 feet vertical, and a water pressure of 131 pounds to the square inch; length of column pipe 342 feet; diameter of column 12 inches; distance from steam-boilers, 2,300 feet.

Year.	Days.	Gallons.
1886.....	329	256,587,148
1887.....	310	197,272,196
1888.....	319	197,532,741
1889.....	360	222,800,422
Totals..	1,318	874,192,507

3,651,904 tons of 2,000 pounds, or = 2,771 tons per 24 hours, or day, or nearly two tons per minute.

"Or equal to a travel for 1,318 days of 84,539,274 feet, a fraction over 16,022 miles, or about twelve and one-sixth ($12\frac{1}{6}$) miles per twenty-four hours. The average for four years is 27,636 gallons of water per hour, but there have been times when we have pumped 67,080 gallons per hour with this pump, and in all that time we have had only one shell leaking. I am satisfied that the expense on the pump during that time has not been ten dollars outside the ordinary wear and tear.

"Yours very respectfully,

"JAMES G. BATEMAN, *Mining Foreman.*

"LYKENS, DAUPHIN CO., PA., *February* 24, 1890."

A little examination of the circulars will show those not acquainted with the mining of anthracite coal the reason that anthracite coal is not bringing a price that will pay for mining and preparing it for market. If they could sell the water that has to be pumped in connection with the mining of the coal, and throw the coal away, they would make money; unfor-

tunately they cannot do that (I make this remark merely to call attention to the quantity of water pumped in proportion to the amount of coal mined).

Mr. Daniel Ashworth—The author of this paper has been very frank and modest in the description of the operations of the series of tests, and it is much to be regretted that they are surrounded by so many assumptions and so much of uncertainty. Undoubtedly it has involved a great deal of labor, together with a great deal of time and attention. Familiar as I am, in my practice, with indicator diagrams, I would say, as far as the steam engines are concerned, that if I were to undertake to remodel such a plant I would take the whole thing out, root and branch, and put in something modern. They bear upon their face a very antiquated type of mechanism, and these diagrams show an exceeding waste to start with. I am aware that there are circumstances and conditions in which it is impossible, as the writer has stated it, to get these down to where we wish them. I think it is to be regretted exceedingly that so much time and so much interest have been manifested in getting up these results, which seem at the end to be full of sound and fury, signifying nothing. I would be very much pleased, indeed, to see at any time a thorough and complete and exhaustive test. in which we would arrive at more accurate results—in which there would be something tangible. But the pumps, I would say, want to be torn out and something modern put in.

Mr. Frederick Meriam Wheeler.—Mr. Allison remarked, as he took his seat, that it hardly pays to mine this anthracite coal, and if they could sell the water and throw away the coal it would be a good investment. It seems to me that if they could improve the economy of these pumps which take so much of their coal, perhaps he might modify his remarks somewhat. Or, in other words, it would not cost them as much to mine the coal. As Mr. Norris remarks, by compounding these pumps they could no doubt save a great deal of money. As a matter of fact, in direct-acting pumps we gain more in compounding a pump than you do in the ordinary steam engine, for well-known reasons. In fact, we expect always to secure a saving of at least one-half. Now imagine half their coal bill saved by simply compounding the pumps.

As regards the economy about which Mr. Norris speaks in his

paper (eighty pounds), that is certainly very flattering. I am inclined to think it is too good a showing. I think the cylinder condensation is very much more than he imagines, especially with the ordinary duplex pump such as they are using. I imagine most of his pumps are of that make. In making some tests on the cruiser *Minneapolis*, not long ago, the economy of the pumps, mostly of the duplex type, varied very much according to their pressure and their speed. For instance, the main feed pumps, running at a fair rate of speed, showed at their best an economy of about ninety pounds weight of steam per indicated horse-power per hour. Those tests were very carefully made, considering that they were at sea at the time, and were done with a surface condenser, measuring quite accurately the water of condensation from the pumps. Of course Mr. Norris could not make any such tests, and they had to approximate the condensation; or, in other words, his economy is simply from figuring the indicator cards. Then, again, clearance loss is very large in the duplex type of pump. Very rarely do you find a pump making its full stroke; or if it does on one side it will run perhaps shorter on the other; consequently the clearance is a very serious matter. Take a pump with say twenty-four inch stroke, and if it is short at one side on one inch, it adds four per cent. to the clearance loss, to say nothing about the normal or ordinary clearance in the cylinders. I would say that the actual amount of steam used by those pumps was at least fifty per cent. higher than he has given in his tables. The slippage he speaks about in the feed pump is certainly very little, and as the water, I understand, was accurately measured in tanks, so we must accept it as quite correct. It certainly is very favorable, especially as you remark the pumps get pretty rough usage. There is no doubt but that, repeating the remark that Mr. Norris made, it would really pay to put in a more economical type of pumps, even if they were the ordinary direct-acting system.

Mr. Henry I. Snell.—In the discussion of this paper, I would like to refer to Table No. 4, on page 57, where a comparison is made between the results obtained in creating blast by a fan blower of very imperfect construction and design, and by steam jets. For information, I would ask some questions in regard to some of the data in this table, which I do not fully understand.

First, the average water pressure obtained by the use of the

steam jets is given as .44 of an inch pressure per square inch, and with the fan blower an average water pressure of .52 of an inch was obtained.

Second, the amount of power in the air delivered by the steam jet is .172 horse-power, and by the fan blower it is .280 horse-power. As the amount of air delivered, multiplied by the velocity due their respective pressures, would be the measure of power, it would follow that the fan was delivering the larger volume, and this is confirmed by the table itself, where it states the volume of air from the steam jet is 2,502 cubic feet per minute, and from the fan 3,506 cubic feet.

Third, the amount of coal burned with the steam jets is given as 962.5 pounds per hour, and only 762.5 pounds with the fan. Apparently, it would seem, the larger amount of coal would be consumed with the larger volume and higher pressure of air, especially when the combustion seems more complete with the fan, as shown in the table by the percentage of ash in the two cases.

Another point, he gives in the same table the percentage of the total horse-power generated by the three boilers that are used by the steam jet and by the fan blower, which, in the case of the steam jet, is 7.4, and of the fan 4 per cent. This table was first published in the *Colliery Engineer*, in answer to an inquiry from one of its correspondents, who was using steam jets under his boiler grates, and who asks: "Could we supply air by fan power with much greater economy, and just as effectually?" While the results stated in the paper may be correct, as applied in this case, with a fan not well adapted to the work, as would seem to be the case from the data given in the paper, yet the facts certainly are misleading in giving the comparative results obtained in innumerable cases in more modern practice and application. In the anthracite regions, using the same kind of refuse fuel, there are many cases where fans using about 18 horse-power to drive them are applied to boilers that are developing 1,800 to 1,950 horse-power, and in a paper read by Mr. Jay M. Whitam before this Society at its St. Louis meeting, he gives the power necessary to operate a fan applied to a battery of boilers developing 410.5 horse-power as 3.21 horse-power.

Taking the case where blast is produced by means of a steam jet, I would cite a test made by Mr. John R. Wagner, a member of this Society, where eighteen plain cylinder boilers, set in

nests of three, each nest having a grate surface of 54 square feet, or a total grate surface of 324 square feet; at the rear of these boilers was a Climax boiler, using only the waste heat from the cylinder boilers. The combined capacity of the cylinder and Climax boilers was 520 horse-power; the amount of rice coal burned per grate, 54 square feet, was 429.3 pounds per hour, the economic evaporation was 5.9 pounds of water per pound of coal under actual conditions, 7.08 pounds of water per pound of coal from and at 212 degrees Fahr., and 10.35 pounds of water per pound of combustible from and at 212 degrees Fahr.

The blast was produced by two steam jets for each of the six furnaces, and the total amount of steam used for the blast was 8.78 per cent. of that generated. A Sturtevant fan was afterwards applied to this plant, and while the rate of combustion was increased about 50 per cent. the power used was not measured, as it was evidently so far inside the limits guaranteed it was unnecessary. An estimate approximately made on the basis of 65 pounds boiler pressure, 100 revolutions per minute of a plain slide valve engine, 9-inch by 12-inch cylinder, and the throttle open only $\frac{1}{4}$ of a turn, could have developed not more than 5 or 6 horse-power. These facts would seem to demonstrate that the average power required to operate a fan should not exceed about 1 per cent. of that developed by the boilers, while the power required to create a blast by the steam jet will be 8 per cent., and sometimes more, and when results like those given in this paper before us, where the relative percentages of the horse-power developed are given as 7.4 to 4, or a saving of about 50 per cent. by the use of a fan, they do not indicate the true relation between the fan and steam jet, which average experience shows should be about 8 to 1 in favor of the fan.

There is another incidental saving in the use of the fan blast over the steam jet, in the less amount of ash to be disposed of, and one superintendent of a coal mine told me that by substituting a fan blower for the steam jets, upon a battery of 24 cylinder boilers, he was enabled to dispense with the labor of one man for hauling away the ashes, which reduced the expense of operating the plant of about three hundred dollars ($300) per year, besides obtaining a better economy in the consumption of coal. I have noted a practice among the collieries of introducing the exhaust steam from the fan engine into the main air pipe

with apparently good results as a means of cooling the grates and to prevent clinkering.

Mr. Allison.—Our friend here speaks of putting in modern pumps and so forth. I would just remark that that would not diminish the quantity of water which would have to be pumped, or the difficulty attending the pumping of mine water containing acids, and the cost of fuel at the mines is about the lowest notch you can get it to. The fuel at the mines costs the minimum price, and no modern pumping machinery which could be put in the Lykens Valley Collieries would change the conditions. It would still remain true, that if they could sell the water instead of the coal they would make money.

Mr. Wheeler.—Less steam means fewer boilers and less labor for attendance. That is an item.

Mr. Ashworth.—I will say this also in connection with what my friend, Mr. Wheeler, has said: Wherever there is waste connected with any operation, while it may be an infinitesimal factor immediately at that point, it becomes contagious through all departments, and you cannot train any of the auxiliary branches to economy if a palpable waste is going on in the principal departments.

Mr. H. N. Suplee.—As regards this matter of putting in economical machinery when the fuel is very inexpensive, I think it must be remembered that fuel is not the only item, as Mr. Wheeler has just stated. I know from my own experience that in a large saw mill, where the fuel was a waste article—saw dust —"offal" as they called it—it paid decidedly to put in economical engines and boilers and install an economical plant simply on account of the increased saving in handling and general maintenance. I think this might be a similar case.

Mr. Henry C. Meyer, Jr.—On Table 9 the author gives the condensation in steam lines, basing the deductions upon a condensation of .012 of a pound of steam for 24 hours per square foot per degree difference in temperature. I find on the bottom of page 65 that that rate of condensation approximates very closely to the results obtained by Professor Norton, which were reported at the Niagara Falls convention. That is a difference of about 20 per cent. between the results of Professor Norton's experiments and those of Mr. Brill, reported at Detroit. I think it was Professor Carpenter who raised the question as to whether the heat transferred from oil through the pipe and

covering, the conditions of the Norton tests, was the same that it was if steam was in the pipe, as in the tests of Mr. Brill. I would like to know whether there are any data as to which is the most reliable figure to use.

*Mr. Norris.**—In replying to the discussion on the paper, it seems proper to first remind you of its original purpose, which was to state as accurately as circumstances permitted the actual condition of the steam plants at these two collieries. We are well aware that the plants are extremely antiquated, and very far from economical, as are those at practically all collieries in the anthracite region. Mr. Allison's statements of the enormous amount of water pumped apply only to the Lykens Valley Colliery; the pumping plant at Summit Branch is still larger and in much worse condition. (See pp. 72 and 75 and Tables 5 and 6.)

It would undoubtedly pay, as Mr. Ashworth suggests, to tear all the machinery out, root and branch, but such a radical course would involve an expenditure of such proportions that it would not be considered for a moment by the directors of any anthracite mining company in the present condition of the business, so all that is left us is to try and do better with what we have. His criticism of the entire work seems hardly justified by the results obtained. While there are some assumptions, clearly stated as such in the paper, a very great effort was made to obtain accurate data throughout, under enormous difficulties, as the inside of a mine is not a favorable place for conducting scientifically accurate experiments. The comparatively small amount of steam unaccounted for in the large Lykens Valley plant (Table 11) seems to indicate that the work was done with at least reasonable accuracy, considering the conditions, and the practical result has been a very large saving in steam. There can be but little doubt that we have underestimated the cylinder condensation in the pumps, but not by any means to such an extent as Mr. Wheeler seems to think; even if all the unaccounted for steam (including all leakage, waste from cylinder cocks, straps, etc.) were to have been used by the Lykens Valley pumps, their steam consumption would have been but 72 pounds. The comparison of these large single mine pumps running at very high plunger speeds with duplex boiler feed pumps is manifestly unfair; they are more nearly comparable with long-

* Author's closure, under the Rules.

stroke slow-speed engines; the clearance is actually very small in percentage, and they all make practically full stroke. I know of no published experiments on the actual cylinder condensation of pumps of this size and type, and being unable to make any, was obliged to use the only available figures from engine practice.

The comparison of fan and steam jet draught was taken, as stated, from the *Colliery Engineer*. The only probable explanation of the discrepancies in quantities pointed out by Mr. Snell is in the character of the coal, which clinkers badly, requiring a large percentage of moisture in the air to keep the fires open with forced draught, and while the exhaust from the engine was sufficient to permit handling of the fires, they were probably much less open than with the steam-jet blast. The fan was, as stated, not at all an economical one, and there is no doubt that a ratio of 7 or 8 to 1 between the economy of the steam jet and fan blast can be readily attained.

Mr. Kent.—Did you ever try injecting a little spray of water into the air to cool the ashes?

Mr. Norris.—Yes.

Mr. Kent.—Isn't that good?

Mr. Norris.—Yes, we do that; but we usually can find enough exhaust steam to do the work, and it seems to last better than water.

Mr. Wheeler.—If you would allow me one minute, I would like to say to Mr. Norris that I did not wish to infer that the cylinder condensation was as high as 50 per cent. But I do wish to say this: that we find in direct-acting pumps it usually runs very high and is even more than 50 per cent under certain conditions. Furthermore, he explained that most of his pumps were single, which I did not know, and of course there this item of cylinder condensation is less. When I referred to the feed pumps of the cruiser *Minneapolis* taking about 90 pounds weight of steam, they were running at a fair rate of speed, feeding the main boilers. A duplex pump of the same size, working on the donkey boilers and running therefore very slowly, took nearly 200 pounds weight of steam. If that increase is not cylinder condensation I would like to know what it is, taking for granted that the strokes were the same in each case and therefore the clearance line—this was per indicated horse-power per hour. I refer you to Chief Engineer W. W. White's paper, published

about a year ago, read before the Society of Naval Architects and Engineers, entitled: "Steam Consumption of the Main and Auxiliary Machinery of the U. S. Cruiser Minneapolis."

Mr. Norris.—Our mine pumps were running at a very high speed, at which pumps, especially single-acting ones, are apt to make nearly full stroke, and owing to the very long average stroke (nearly 6 feet) the clearance is exceedingly small. We found the consumption of steam in our boiler feed pumps enormous at times, much exceeding that in the mine pumps.

Mr. Wheeler.—I considered that a very favorable showing for those pumps, even allowing the moderate clearance loss.

Mr. Norris.—As regards the condensation losses referred to by Mr. Meyer, it will be noted that we made two tests (page 65), the results of which agreed almost exactly with those obtained by Mr. Brill; these two, however, were made on pipe with covering better than our average condition, and for our average covering the figure 0.012 pound condensation in 24 hours per square foot for 1 degree Fahr. difference in temperature (the figure obtained by Mr. Norton) was considered safer. Our coverings are principally asbestos.

Mr. Ashworth's remarks as to the contagion of waste in any department hardly apply to the steam plant of anthracite collieries. The waste, while very large, is not palpable to the ordinary miner or laborer, and it does not seem just to cast such a broad aspersion upon the management of anthracite mining, which, except in this one particular, is almost universally notably efficient and economical, especially as the coal used for steam purposes was, up to a few years ago, practically a waste product, and the steam plants, as they are now, have developed under that condition.

DCCXCV.*

IMPROVEMENTS IN STEAM BOILERS, AND THE BRICK SETTINGS OF SAME.

BY WILLIAM BARNET LE VAN, PHILADELPHIA, PA.

(Member of the Society.)

THIS boiler herewith presented is of the horizontal-flue class; and its object is to afford, in connection with structural strength and simplicity, effective and desirable means for the generation of high-pressure steam, which in operation shall present the advantages of freedom from foaming, quick steaming capacity and production of superheated steam, ample releasing surface and steam room, thorough utilization of heat and combustion of the gases evolved from the fuel, and facility of removal of scale or deposit of solid matter.

The writer, after an experience of over forty years, is satisfied that it is the best form of boiler which can be erected, considering all conditions. At the present time the desire of most boiler-makers is to secure accuracy and solidity of workmanship, which will defy for a long period the continual strain due to the high steam pressures now carried to produce economy in fuel.

The improved horizontal boiler is shown in Fig. 22 in longitudinal vertical section upon the line *dd* of Fig. 23, and is so arranged that the conducting power of the metal removes the heat from its inner surface by the perfect circulation of the contained water due to the slight incline of the main shell from front to rear, so as to put the largest body of water at the back end, which is located in the hottest part of the furnace when fired with a flaming fuel; the flues being so arranged with a large water space on the bottom, and ample waterways on each side between flues and shell, as not to impede the fall of water to the bottom of the boiler.

It is well known that water, on account of its capacity for

* Presented at the New York meeting (November, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

heat, should have its spaces within the boiler so arranged as to give it free access to the surface that receives the greatest heat. The well-known law, deduced from experiments, is that a given

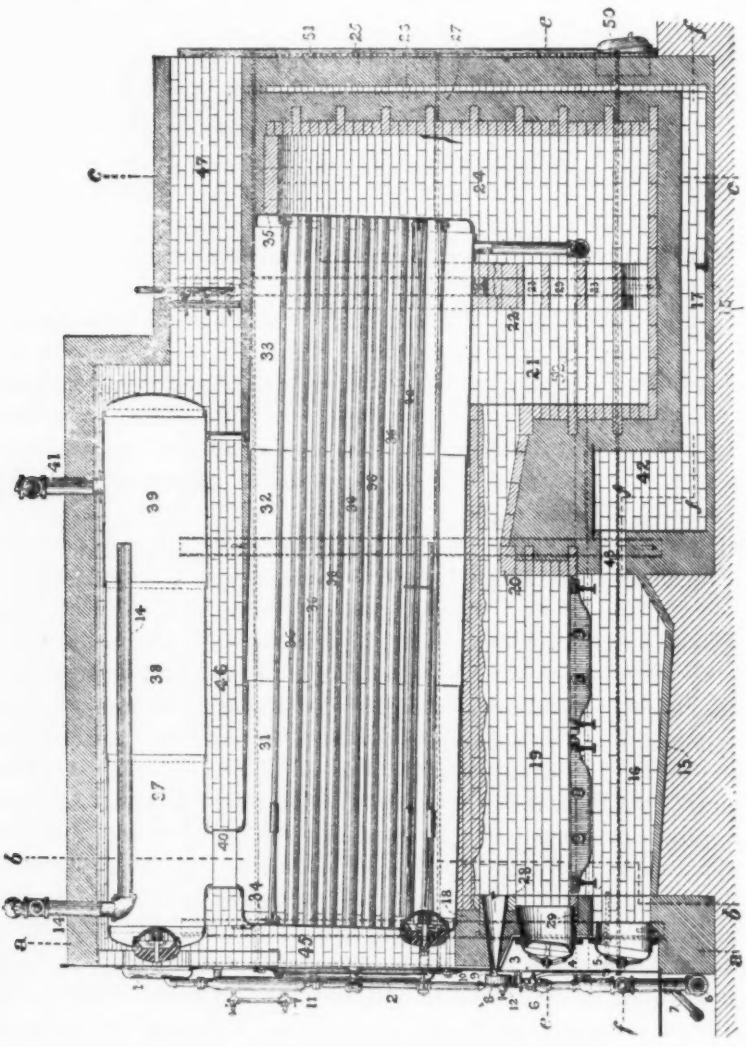


FIG. 22.

area of plate surface, after its temperature has become constant, will transmit the entire number of units of heat received, if the recipient be water. If the recipient be steam, it will transmit only 60 per cent. Therefore, the free and adequate means of cir-

culation of the water and unobstructed access to the plates are the advantages attained in this boiler.

It is a well-known fact that steam cannot ordinarily be superheated when in contact with the water from which it is generated. I therefore have isolated the steam from the water by making the steam drum of large dimensions and connecting it to the main shell of the boiler by a single neck located at the front end of the boiler and taking the steam from the rear end of the superheating steam drum, thus allowing time for the surrounding highly heated gases to evaporate any entrained water or foam in the steam drum before leaving the boiler.

Fig. 23 shows the duplex fire front with cleaning door to steam drum, and flue doors, fire and ash-pit doors, feed and blow-off pipes, water columns, and cold-air registers for supplying hot air into the ash pit when ash-pit doors are closed.

Fig. 24 shows a horizontal half-section through the line *cc* of Fig. 22. Fig. 25 is a similar section through the line *ff* of same figure, showing plan of foundation of the boiler setting. Figs. 26, 27, and 28 are vertical half-sections upon the lines *aa*, *bb*, and *cc* of Fig. 22, respectively. Fig. 29 is a half rear view of the back of boiler setting.

The improved boiler is a cylindrical shell composed of a series of rims or plates, 31, 32, and 33, which are riveted one to another in telescopic form successively, and continuously increasing in diameter from the front head 34 to the back head 35. The usual flues extend from the front to the back end, in the usual manner, and are preferably so arranged as to leave sufficient clear space at the bottom of the boiler for a man to enter underneath the flues, and also leaving space between the fire flues and the shell of the boiler along the sides of the latter, so as to allow a free circulation of water along the hottest portion of the boiler shell from the top to the bottom of the same.

A cylindrical superheating steam drum, 37, 38, 39, is connected to the main shell of the boiler at a point adjacent to its front end by a single neck, 40, so that the drum shall, when the boiler is in working position, be horizontal, or substantially so, while the main shell of the boiler rakes downwardly towards its rear and larger end, 35. The superheating steam drum is constructed telescopically in a manner analogous to the main shell, its rear rim, 39, being the largest in diameter. The steam outlets, 14 and 41, are connected to this steam drum, the safety

valve's outlet, 14, being extended on the inside of the drum rearwardly as far back as the rim, 39, in order to secure absolutely dry superheated steam.

The transverse joints of shells are single riveted, longitudinal seams triple riveted butt joints, with inside and outside covering strips. All the longitudinal joints are placed above the

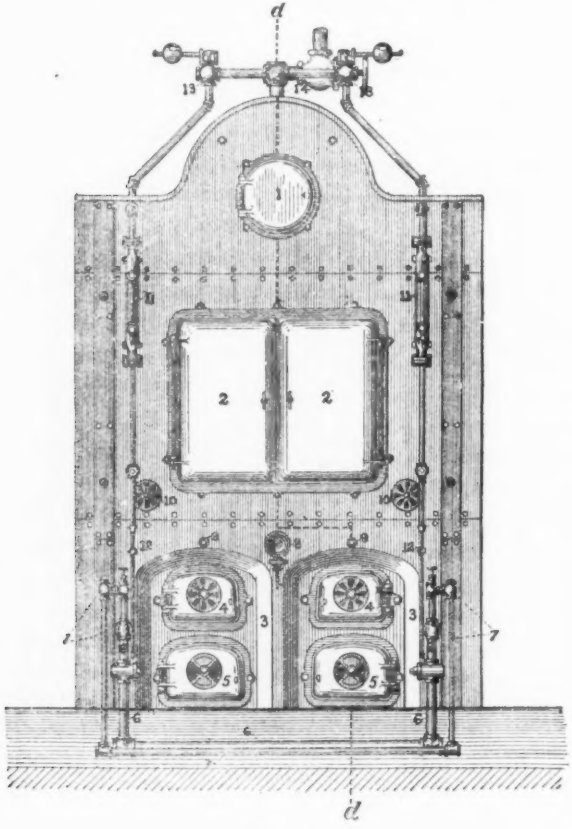


FIG. 23.

water line. Heads are well stayed with angle iron and braced by 1⅛-inch rods one to the other, thus releasing transverse joints from longitudinal strain. The braces are fitted with "pipe swivels" or turnbuckle, with screw ends cut right and left handed so as to equalize the strain on the heads The flues are well expanded into and beaded over end plates, and so arranged that

there is a clear space vertically of 3 inches between shell and flues, to allow a free circulation of the water.

The neck connecting the superheating steam drum to the

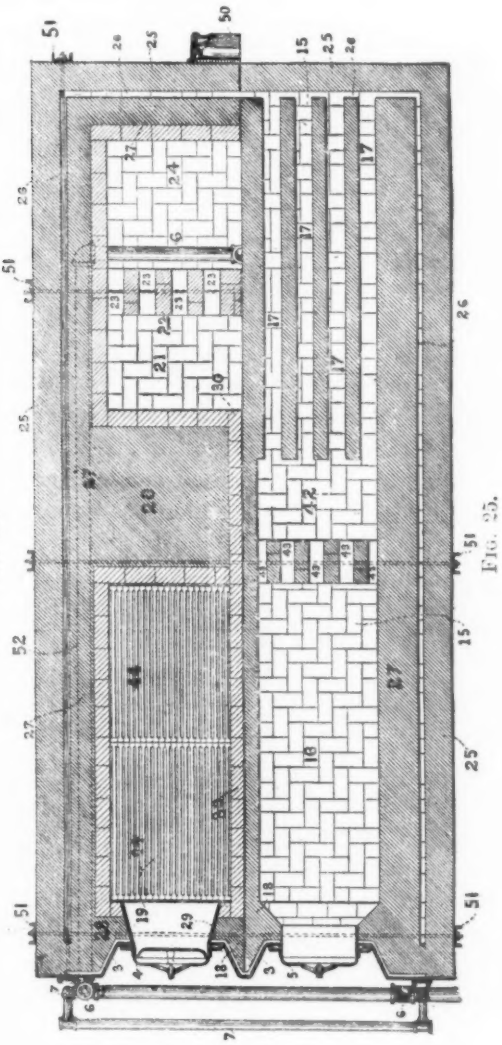


FIG. 34. FIG. 35.

main shell should be at least 16 inches internal diameter. To reënforce the above, due to the large opening for the neck, the openings are cut small in the former and latter, and are each flanged outwardly to fit the neck and riveted to the same,

thereby reducing the weakening of the shells caused by the removal of metal.

All the rivet holes are punched $\frac{3}{32}$ of an inch less in diameter than the rivet; and when all the plates are brought well together in place by temporary bolts, the holes are reamed fair to receive the rivets and countersunk slightly so as to form a fillet to rivet head, the countersinking being done by a steel pin, so that there is no reduction in value of the plates—as would be the case if cut away by a reamer.

Manholes, 11 x 15 inches, oval or an ellipsis in form, are placed in the front heads.

All the shell plates are best *mild steel*, and do not exceed 60,000 pounds per square inch tensile strength, with an elongation of 30 to 50 per cent. in specimens 1 inch wide and 2 inches long, and a reduction of area 35 to 60 per cent. All plates are also subject to the *temper test*, which consists of heating a strip of plate, say 2 inches wide and 15 inches long, to a bright red and plunging it into water of about 80 degrees temperature and then bending it cold. The range and suddenness of the change of temperature in cooling, and the degree of consequent bending of the plate, will serve to indicate its quality with respect to ductility and soundness.

As before stated, all horizontal seams are *double welt* butt joints, triple riveted. Thus the shearing of the rivets must occur in three places; and on this account their resistance is very nearly twice as great as in other joints. This joint is free from the distortion, on account of the oblique action of the stress on the rivets, to which the lap joints and single welt butt joints are subject. These butt joints distribute the strain at the joint uniformly over the whole section of the metal, whereas with an ordinary lap joint the strain is concentrated at the edges of the overlapping plates. All transverse joints are shrunken together, and the rivet holes reamed to rivet size and slightly countersunk.

Improper Boiler Settings Augment the Production of Smoke.

The real cause of the formation of smoke from our power plants is due to the absence of the proper supply of air for the combustion of the gas (the only combustible that it contains)

at the time when, from its high temperature of incandescence, it is best fitted to receive it.

Up to the present time boiler engineers have not given due

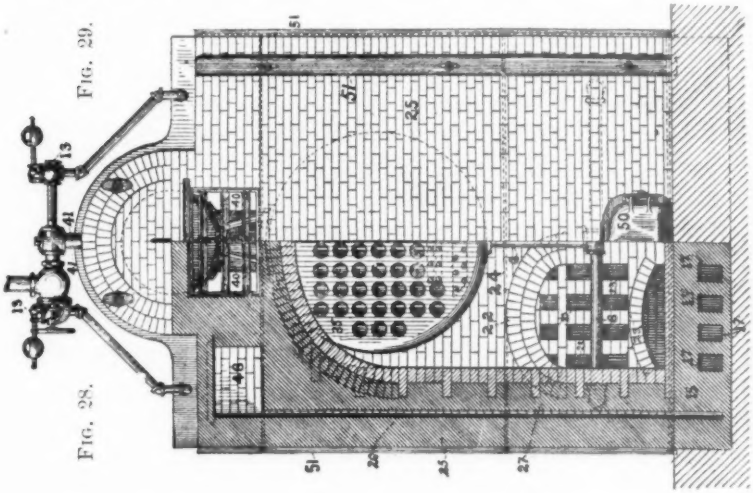


FIG. 29.

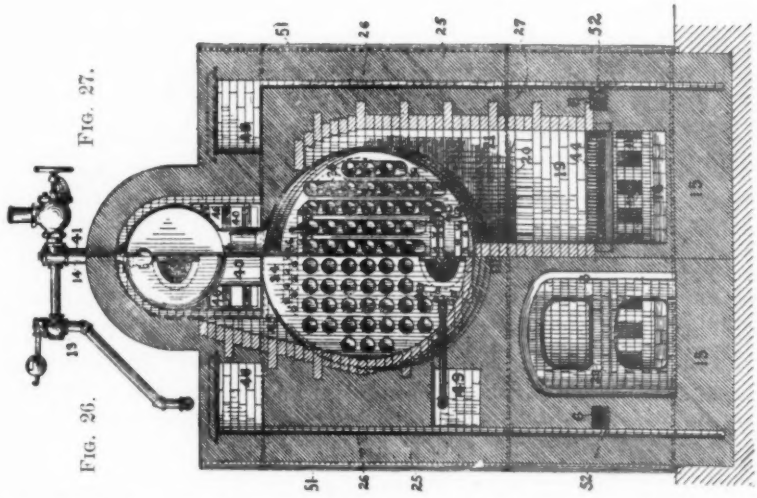


FIG. 28.

FIG. 27.

FIG. 26.

consideration to the providing for the admission of this supply of air to the furnace at the proper time and in the proper quantity, it being considered of secondary importance, and in a majority of cases is wholly neglected.

C. Wye Williams says the idea is palpably erroneous that

smoke once formed can be consumed in the furnace in which it is generated. The formation of smoke, in fact, arises out of the failure of some of the processes preparatory to combustion, or the absence of some one of the conditions which are essential to that consummation from which light and heat are obtained. To expect, then, that smoke, which is the very result of a deficient supply of heat or air, or both, can be consumed in the furnace in which such deficient supply has occurred, is a manifest absurdity, seeing that if such heat and air had been supplied this smoke would not have existed. Smoke is the unconsumed portion of black carbon; insignificant though it may be in weight or volume, this carbon of the gas is the sole black-coloring element of smoke. Flame is not the combustion of the gas. Flame itself has to undergo a further process of combustion, being but a mass of carbon atoms still unconsumed, though at the temperature of incandescence and high luminosity. Flame is, then, but one of the stages of the process of combustion. Its existence marks the moment, as regards each atom of its separation from and the combustion of its accompanying hydrogen, by which so intense a heat is produced as instantaneously to raise the solid carbon atom, then in contact, to that high temperature; thus preparing it the more rapidly to combine with oxygen so soon as it shall have obtained contact with the air, but not a moment sooner.

As boilers are commonly erected, instead of admitting the air while the carbon in the fire chamber is at its highest temperature, the custom is first to allow the carbon particles, or even force them, to cool down, by contact with the metallic sides and flues or tubes, to the state of soot; and then expect, by some mechanical apparatus, to restore them to the necessary temperature from which they had been so gratuitously reduced. In the improved boiler setting, as will be explained further on, the air is admitted to the carbon over the fire grate before the high temperature of the carbon is reduced by coming in contact with the flues.

The first thing requisite in a proper boiler setting is to provide for the admission of the quantity of air chemically required for the perfect combustion of a given quantity of coal, of the quality commonly used for steam purposes. The amount of air to each pound of coal has been determined by experiments to be, in round numbers, 150 cubic feet, of which 45 cubic feet are

required for the various carburetted hydrogen gases given off, and 105 for the solid carbon. The practical utility of this knowledge, however, is much impaired by the circumstance that combustion ceases even in pure oxygen, and much more so in air, before the whole of the oxygen present has entered into the new chemical combinations required. It is also known that carbonic acid gas exerts a positive influence in checking combustion, as a candle will not burn in a mixture composed of four measures of air and one measure of carbonic acid gas.

Large quantities of this gas are generated by the combustion of the solid carbon on the grate, and are necessarily mechanically mixed with the inflammable gases as they rise. The quantity of air required for their subsequent combustion, therefore, must be increased to a very large extent. The whole of the air thus supplied in excess must be heated to a very high temperature before any combustion can take place, and the loss of heat thus absorbed must be taken into account in calculating the ultimate economy of igniting these gases.

In the combustion of the solid portion of coal, as left in an incandescent state on the fire grates of a common furnace, after the volatile gases have passed off, the amount of heat generated by the whole of the carbon, uniting at once with its full amount of oxygen, will be the same as would be generated by its combination, first, with a smaller quantity of oxygen, forming carbonic oxide, and, subsequently, by the ignition of this gas, by its combination with the further quantity of oxygen required to turn it into carbonic acid gas.

As some portion of the carbon is always converted into carbonic acid gas in the furnace, it follows that the air for the ignition of any carbonic oxide there formed and allowed to pass into the flues must be greatly in excess of the quantity chemically required; and the whole of this excess must be raised to the temperature of the other gases with which it will commingle.

The superior economy, therefore, of at once converting the whole of the carbon into carbonic acid gas is apparent; and there is no doubt that this very desirable result may be obtained nearly to the full extent by due care in the formation and subsequent management of the furnace.

The best mode of supplying air to the other inflammable gases resulting from the combustion of the fuel, which are composed of hydrogen and carbon, besides that which is passed through

the grate-bars of the furnace, has been a matter of much controversy, and been the subject of many patents, but it has been found by a large number of experiments that the proper place to introduce it is through the front of the furnace, including the fire door, the latter being constructed hollow, with an inside lining perforated with a large number of small holes and a register on the outside of the door that can be regulated by the fireman to admit the exact quantity required, according to the kind of fuel used, by actual observation through the sight or "peep holes," 9 and 9, covered with glass, and situated directly over the furnace door. The additional air passed through the furnace door is heated by the thin liners.

As the carburetted hydrogen gases are generated rapidly on the application of heat to the coal, and are in themselves much lighter than carbonic acid gas, or the nitrogen gas, formed at the same time, it is sometimes assumed that they rise nearly unmixed to the top of the space over the fire grates.

Therefore, the boiler should be set from 30 to 40 inches above the grate bars, which allows ample room for the air admitted through the furnace doors and fire fronts, and the gases to mix and combine. To accomplish this thoroughly, the products of combustion are partially delayed by a "septum" or division wall, 22, Figs. 22 and 27, which separates the fire grate and combustion chamber, 21, from the back end of the boiler where the gases enter the flues, and by the interposition of the septum wall the gases of combustion are retained in prolonged contact also with the wetted surface of the shell. This division wall extends up to and completely around the boiler, causing an upward and a downward course of the current of hot gases, thus insuring a better combustion and positively assuring the thorough commingling of the fuel gases and the atmospheric oxygen by expanding and contracting their volume from one chamber to the other, thus compelling them to come in contact with the highly heated fire-brick division wall, 22, and forming flame, which cannot exist below a temperature of at least 800 degrees Fahr.

By this means greater efficiency is obtained by reason of the slight retardation in the velocity of the gases of combustion in their exit from the combustion chamber.

The combustion chamber over the grate bars is continued up to the water line, leaving a space between the boiler and com-

bustion chamber walls of 6 inches, thereby allowing the products of combustion to impinge on all the *wetted* surface before entering the flues at its highest temperature, thus securing absorption at the greatest heat, while the balance is passed through the flues and around the steam drum into the chimney at a temperature which renders the use of a separate economizer unnecessary.

Incomplete combustion of fuel in the ordinary boiler setting is due to the shallowness of the combustion chamber over the fire grates, and want of proper air supply, causing the chilling and extinguishing of flame through contact with the surface of the boiler before the combustion is completed. This is entirely obviated in this improved boiler setting, as can be seen through the observation holes in the fire front.

The injurious effect of leaving too little room *between the fire-grates and the boiler*, thereby refrigerating the flame, may be illustrated by a very simple experiment. If a small copper or porcelain capsule containing water be held over the flame of a gas burner, a little above its apex, the flame will suffer no abatement of brightness or size, but will continue to keep the water briskly boiling. If the capsule be now lowered *into the middle of the flame*, the latter will immediately lose its brightness, becoming dull and smoky, covering the bottom of the capsule with soot, and owing to the imperfect combustion, although the water is now surrounded by flame, its ebullition will cease.

It is a well-known fact that metal flues act as extinguishers of flame; therefore, to produce the best results, it is necessary that the combustion of the gases evolved from the fuel be entirely completed before they reach the flues, and this is accomplished, in conjunction with the "septum" wall, by making a duplex fire chamber the depth of the fire grate.

In my practice I have found a considerable improvement was effected by constructing duplex furnaces. It has the important advantage of rendering any interference with the supply of air unnecessary, by giving uniformity to the quantity of gas from the bridge wall to the flues; since, by firing the two furnaces *alternately*, the supply of gas is equalized on entering the flues.

By the use of duplex fire chambers a very important result is gained, consisting in the fact that the heat of the gases evolved from the fuel in the fire chambers is absorbed by the division wall between the two furnaces, and while cleaning the

grate bars of one of the chambers from ashes and clinker, etc., the steam pressure in the boiler is not sensibly reduced, due to the stored heat given out by the highly heated division wall, the same being the case in putting on fresh fuel. The furnace can also be fired alternately, thus maintaining a regular steam pressure in the boiler.

Some engineers entertain the opinion that a sufficient supply of air for the gases may be obtained through the grate bars; and it is obvious that a partial supply, at least, may be obtained in this manner by a judicious management of the fire. This may be effected by keeping the fires thin and open, and feeding the fuel in small quantities at a time. The air which passes through is not vitiated further than in being mechanically mixed with the carbonic acid and nitrogen gases, caused by the combustion of the coal on the grate bars.

The perfect combustion of the whole ingredients of coal being entirely dependent, chemically considered, on the supply of the quantity of atmospheric air, it is evident that the velocity with which the air flows into the fire will materially affect the result. According as the velocity is greater or less, so in proportion must the quantity of coal that is to be consumed on a given area of grate be increased or diminished, and there is no limit to the quantity that may be so consumed, beyond the difficulty of supplying the air sufficiently rapidly. The various circumstances which affect the velocity of the entering air have placed this question, as yet, completely beyond the reach of theory, so that practical experiments must be taken as the only guide in determining what quantity of air can be made to enter into a given furnace, and, consequently, what amount of coal can be properly consumed in a given time.

The air, in my improvement, for the supply of the fire is not taken wholly through the ash pit doors. These latter are kept almost shut—in fact, at times, entirely shut. The air supply is delivered into the ash pit, not in a *cold* stream, but *hot*. A double air flue, 26, Fig. 25, is formed in the outer casing on each side of the boiler, as well as around the back end, and also under the bridge wall there is formed a hot-air chamber, 42, which receives all the hot air which radiates from the inner walls. By this arrangement the air entering the inlet registers, 10, 10, Fig. 23, located in the fire front, passes the whole length of the side walls, and is delivered into the hot-air chamber,

under the bridge wall, and from there it is delivered under the grate bars at a high temperature, having, in passing amongst the hot bricks, become considerably increased in volume, due to its absorption of heat, so that it becomes an actual hot blast through the bed of coals, resulting in a very perfect combustion of the fuel.

The advantage of considerable amount of space in the furnace over the fire grates has been already mentioned, but no very decisive experiments have been made on this subject. Three cubic feet of space to each superficial foot of grate bar surface may be stated as a good proportion where there is nothing to prevent this amount being obtained. When the space is reduced below one foot and a half of grate it will be found to be attended with a marked disadvantage.

The area of the flue, and subsequently of the chimney, through which the products of combustion must pass off, must be regulated by the bulk and the velocity of the latter. The quantity of air chemically required for combustion of 1 pound of coal has been shown, as stated, to be in round numbers 150 cubic feet, of which 45 enter into combustion with the gases, and 105 with the solid portion of the coal. From the chemical changes which take place in the combination of the hydrogen with oxygen the bulk of the products is found to be, to the bulk of the atmospheric air required to furnish the oxygen, as 10 to 11. The amount is, therefore, about 49 in round numbers. This is without taking into account the augmentation of the bulk, due to increase of the temperature.

In the combination which takes place between the carbon and the oxygen the resultant gases (carbonic acid gas and nitrogen gas) are of exactly the same bulk as the amount of air, that is, 105 cubic feet, exclusive, as before, of the augmentation of bulk from the increase of temperature. The total amount of the products of combustion in a cool state would therefore be:

$$49 + 105 = 154 \text{ cubic feet.}$$

The general average temperature of a boiler furnace has not been very satisfactorily ascertained, but it may be assumed at about 1,200 degrees Fahr., and at this temperature the products of combustion would be increased, according to the laws of expansion of aeriform bodies, to about three times their original

bulk. The bulk, therefore, of the products of combustion, which must pass of, must be as follows:

$$154 \times 3 = 462 \text{ cubic feet.}$$

At a velocity of 40 feet per second, the area, to allow this quantity to pass off in an hour, is 0.52 square inch. In a boiler furnace in which 12 pounds of coal are burnt on a square foot of grate per hour the area to every square foot of grate would be:

$$0.52 \times 12 = 6.24 \text{ square inches;}$$

and the proportion to each foot of grate, if the rate of combustion be higher or lower than 12 pounds, may be found in the same way. This area having been obtained on the supposition that no more air is admitted than the quantity chemically required, and that the combustion is complete and perfect in the furnace, it is evident that this area must be much increased in practice where we know these conditions are not fulfilled, but that a large surplus quantity of air is always admitted. A limit is thus found for the area over the bridge, or the area of the flue immediately behind the furnace, below which it must not be decreased, or the volume of gas could not pass off, and consequently the proper quantity of air could not enter and the combustion would be proportionally imperfect. It will be found advantageous in practice to make the area 2 square inches instead of 0.52 square inch.

The imperfection of the combustion in any furnace, where it is less than one and one-half (1.5) square inches, will be rendered very apparent by the quantity of carbon which will rise unconsumed along with the hydrogen gas. This would give 24 square inches of area over the bridge wall to every square foot of grate in a furnace in which the rate of combustion is 12 pounds of coal on each square foot per hour, and so in proportion for any other rate.

Taking this area as the proportion for the products of combustion immediately on their leaving the furnace, it may be gradually reduced as it approaches the chimney on account of the reduction in the temperature, and consequently in the bulk of the gases. Care must, however, be taken that the flues are nowhere so contracted, nor so constructed, as to cause, by sharp

bends, or in any other way, any obstruction to the draft, otherwise similar bad consequences will ensue.

The septum, or perforated wall, partially retards the current of flame, or hot gases, by reason of such gases impinging against such septum, the result being that the stratum of gases in immediate contact with the wall is first retarded in the speed of its flow, and this results in a measurable retardation of the entire stream, with the final result that an intermingling of all the gases is effected, and part of the heat of all parts of the stream is imparted to the septum until the latter becomes incandescent, after which its heat aids in effecting the perfect combustion of the gases.

Another very prevalent practice, and which also would seem to be open to serious objections, is that the flues formed of brickwork are frequently made of much greater area in one part than in another. The effect of this is evidently to cause a very slow and languid current in the larger part of the flue.

The proper amount of surface that ought to be given in a flue to carry off the heat, or to cool down a given quantity of heated gases, has not yet been investigated with any degree of accuracy, and practice varies widely under different circumstances. At the present writing the usual amount of heating surface is 30 square feet to 1 of grate.

The time occupied by the gases in passing through the boiler from the instant of their generation to that of their leaving the boiler, and the length of the course through which they have travelled, have sometimes been looked upon as matters of great importance. Where the gases are travelling in one compact mass it is evident that distance, and consequently time (as the velocity with which the current flows is the same in all cases), must be allowed for the different particles of this large mass so to circulate among themselves as that each may have an opportunity of coming into contact with a cooling medium, to give off its heat; but if the large mass of gases is so subdivided that the different particles are sooner brought into contact with the due amount of cooling medium, then the time the gases remain in contact with the boiler ceases to be of importance. When the gases have reached the end of a well-proportioned boiler and brick setting they should be reduced to a temperature not exceeding 450 degrees Fahr., or below it; their bulk will in consequence be reduced by about one-third below their bulk on

their first leaving the furnace. The reduction in the area of the flue ought not to be in the same proportion, because their velocity is no longer so great. The reduction ought to be made gradually, as has been stated before, and not by a sudden contraction at the end of the boiler, as the effect of this is to cause a slowness of draft in the latter part of the flue.

The feed water is supplied to the boiler through the blow-off pipe, which is of ample dimensions, it being encased in the side wall of the boiler from the back end of the boiler to the front end, and thus absorbs heat, forming a heated passage for incoming feed water, the result being that the water is delivered to the boiler at nearly the boiling temperature, and also, by reason of such arrangement, the blow-off pipe is kept free from sediment, which otherwise would tend to clog it.

The boiler proper is held in position by a cradle formed of fire brick. The front end is supported by recesses, 3, 3, Figs. 24, 25, and 27, which form a pier, 18, and the division wall, 30, which separates the fire chambers; the rear end is supported by the septum wall, 22, being thus independent of the inner walls, 27, 27, with which it does not come in contact. By this arrangement cast-iron brackets or lugs, such as are usually attached to the main shell of horizontal boilers for the purpose of supporting them in place, are dispensed with.

The fire chambers, 19, above the grate bars, 44, are dual, being separated longitudinally by the vertical wall, 30, which wall ends at or beyond the bridge wall, as may be deemed advisable.

For the accumulation of heated air and products of combustion, which might otherwise leak out and escape, I provide the lateral chambers, 48, on each side of the superheating drum chamber, 46, and extending the entire length of the boiler and communicating with the air-space, 26, and they can be supplied, if so desired, with a surplus of air from the registers, 10, 10, and inlet-space, 49.

In the rear exterior wall of the setting I build a cleaning door, 50, to give access to the back connection chambers and flues for cleaning or repairing purposes.

In the operation of this boiler and settings the heated gases evolved from the fuel are applied to the entire exterior of the wetted surface of the boiler by reason of the upward extension of the fire chambers, 19, whence they pass into the similarly upwardly extended supplemental mingling combustion chamber,

21, to or about the water level. The division or septum wall, 22, deflects the gases to the lower portion of the chamber, 21, and retards their flow. Said septum wall becoming highly heated, in fact incandescent, correspondingly promotes combustion of the highly heated gases; in other words, it acts as a regenerator to promote combustion of the uncombined gases on their way to the rear of the boiler. The gases evolved from the fuel are thus effectually mingled with the proper supply of heated oxygen conducted through the registers, 10, 10, air passages, 26, 17, and 43, into the ash pit, 16, and that introduced through the registers located in the fire doors, 4, 4, and the gases are consumed before reaching the back connection chamber, 24, the resulting products being delivered, consequently, into said chamber in a very highly heated state, due to the retardation by and contact with the incandescent septum wall, 22, causing a thorough mixture of the air introduced through the registers, 10, 10, and the carbonic oxide (CO), under which conditions a second combustion takes place, the carbonic oxide taking an additional part of oxygen, forming carbonic acid (CO_2) with great evolution of heat, as hereinbefore mentioned. The superheated products of this second combustion then pass into the flues of the boiler, 36, 36, 36, etc., at the rear end, 35, of the main shell and through the same to the front flue chamber, 45, thence to superheating drum chamber, 46, and flue, 47, leading to chimney.

A very important advantage of this setting consists in the fact that the gases evolved from the fuel in the fire chambers, 19, are maintained in temperature by the enclosed highly heated walls even during clearing the fire from ashes, clinker, etc. Thus, steam pressure in the boiler is not sensibly reduced during such cleansing operation.

The claim for internally fired boilers that there is no loss of heat by radiation is not borne out in practice, and they labor under the disadvantage of being restricted in their fire box, and are unable to burn to an advantage the cheaper grades of fuel which require large combustion chambers. The metal walls of the majority of internally fired boilers refrigerate the flame, thereby lowering the calorific value of the fuel used.

The advantages of this boiler and setting are shown in an evaporative test on two 54-inch by 17-feet horizontal-flue boilers into single furnace, made by George H. Barrus, Septem-

ber 29 to October 6, 1888, the object of which was to determine the general economy of the boilers and the relative performance of different kinds of fuel, conducted at the Keystone Watch Case Company, Philadelphia, Pa.

TABLE NO. 1.

DIMENSIONS OF LE VAN'S HORIZONTAL-FLUE BOILERS: AGE, SIX WEEKS; HORSE-POWER, ON A BASIS OF 12 SQUARE FEET, SEVENTY-EIGHT EACH.

No.	Description.	
1.	Number of boilers	2.
2.	Diameter of main shell, each, in inches	54.
3.	Length of main shell and flues in feet	17.
4.	Length of drum, each, in feet	12.
5.	Diameter of drums, each, in inches	3.
6.	Number of four-inch flues, outside diameter in each boiler	45.
7.	Pitch or rake of main shell, each, in inches	6.
8.	Size of grate, each, length in feet	8.
	and width in feet	4.
9.	Width of air spaces in inches	0.375
10.	Width of bars in inches	0.625
11.	Area of perforations through fire door in square inches	4.7
12.	Distance of grate to shell, front end, in feet	3.
13.	Distance of top of flat bridge to shell, each, in feet	1.
14.	Kind of brick setting: Le Van's, with air passages in walls leading to ash pit.	
15.	Number and diameter of steam nozzles, each 2	4.
16.	Distance from bottom of glass to top of lower shell, in each boiler, in inches	12.
17.	Area of grate surface, each, in square feet	32.
18.	Area of water-heating surface, each, in square feet	934.
19.	Area of steam-heating surface, each, in square feet	107.
20.	Total area of heating surface, each, in square feet	1.041
21.	Area through flues, each, in square feet	3.4
22.	Area through damper, each, in square feet	2.3
23.	Height of iron stack, 72 inches in diameter, lined with four-inch brick, in feet	120.
24.	Ratio of water-heating surface to grate surface	29.2 to 1
25.	Ratio of steam-heating surface to grate surface	3.3 to 1
26.	Total ratio of heating surface to grate surface	32.5 to 1
27.	Ratio of grate to flue area	9.3 to 1

COLLECTIVE QUANTITIES.

28.	Ratio of surface of grate, two boilers, in square feet	64.
29.	Ratio of water-heating surface, two boilers, in square feet	1.868

TABLE NO. 2.

DATA AND RESULTS OF EVAPORATIVE TEST OF TWO LE VAN BOILERS AT THE KEYSTONE WATCH CASE COMPANY'S FACTORY.

KIND OF COAL USED.	LYKEN'S VALLEY BUCKWHEAT.	FOUR PARTS OF NO. 2 PEA AND ONE OF CUMBERLAND.	GEORGE'S CREEK CUMBERLAND.	LOWRY'S PEA SCHUYLKILL.	GIGG HOUSE WASHED PEA.
Date—1888.	Sept. 29th.	October 1st.	October 2d.	October 3d.	October 6th.
TOTAL QUANTITIES.					
1. Duration of trial in hours	10.2	11.3	11.1	11.	10.
2. Weight of wood used in lighting fires, in pounds	185.	231.	300.	225.	283.
3. Equivalent value of wood expressed in terms of coal, in pounds	74.	92.	120.	90.	113.
4. Weight of moist coal consumed, including wood equivalent and including slight waste in ashes, in pounds	7,079.5	7,635.	6,489.	8,110.	7,289.
5. Weight of ashes and clinkers and waste coal, in pounds	1,452.	1,427.	507.	1,442.	1,313.
6. Percentage of moisture in coal	2.	3.	3.	0.	2.7
7. Weight of combustible consumed (dry) in pounds	5,487.5	5,982.	5,791.	6,667.	5,782.
8. Percentage of ashes and clinkers and waste coal	20.5	18.7	7.8	17.8	18.
9. Weight of water evaporated in pounds	59,765.	64,530.	63,112.	70,631.	62,547.
HOURLY QUANTITIES.					
10. Moist coal consumed per hour in pounds	696.8	575.7	584.6	737.3	728.9
11. Coal consumed per hour per square foot of grate, in pounds	10.9	10.6	9.1	11.5	11.4
12. Water evaporated per hour in pounds	5,882.	5,709.	5,674.4	6,421.7	6,228.
13. Equivalent evaporation per hour, feed water 100 degrees, steam pressure 70 pounds, in pounds	5,346.7	5,184.4	5,158.0	5,837.3	5,661.2
14. Horse-power developed on basis of 30 pounds of water from 100 degrees, at 70 pounds	178.2	172.8	171.9	194.6	188.7
15. Equivalent evaporation per square foot of water-heating surface per hour, in pounds	2.9	2.8	2.8	3.1	3.0
AVERAGES OF OBSERVATION.					
16. Average steam pressure in boiler, per square inch	87.3	81.3	86.3	85.5	86.4
17. Average temperature of feed water, in degrees Fahrenheit	203.	204.	204.	204.	204.
18. Average temperature of flue gases, left-hand boiler, in degrees Fahrenheit	388.	426.	450.	420.	430.
19. Average draft suction, left-hand boiler, height of column of water, in inches	0.16	0.2	0.1	0.24	0.27
20. Weather and outside temperature in degrees Fahrenheit	65 Clear.	65 Cloudy. A. M. Cloudy. P. M. Clear.	65 Fair.	60 Clear.	60 Rain.
RESULTS.					
21. Water evaporated per pound of moist coal, in pounds	8.441	8.452	9.726	8.709	8.581
22. Equivalent evaporation per pound of dry coal, from and at 212 degrees, in pounds	8.999	9.101	10.472	9.10	9.213
23. Equivalent evaporation per pound of combustible, from and at 212 degrees (with dry coal), in pounds	11.381	11.272	11.388	11.071	11.305
24. Tons of coal per day for 65,000 pounds of steam, in tons	3.43	3.43	2.98	3.34	3.38
25. Price per ton, including 50 cents for carting, in dollars	3.15	3.05	3.65	3.25	2.90
26. Cost of coal for a day's run of 65,000 pounds of steam, in dollars	10.78	10.46	10.89	10.82	9.80
27. Cost of removing ashes, at 50 cents for 1,700 pounds, in fraction of a dollar	0.46	0.42	0.15	0.39	0.40
28. Total cost of fuel and removing ashes for one day's run, in dollars	11.24	10.88	11.04	11.21	10.20

NOTE.—The quantities given for water evaporated are uncorrected for the condition of the steam, whether moist or superheated.

Review of Results.

General Economy of the Boilers.

"The degree of economy attained by the boilers is indicated by the evaporations per pound of combustible from and at 212 degrees, given in line 23 of Table No. 2.

"I consider that any result exceeding *eleven pounds* is excellent when obtained with anthracite coal, more especially when it applies to the small sizes of coal such as were used on most of these tests. The highest result which I have obtained with anthracite coal is about eleven and one-half pounds, but this occurred with the broken size, which is usually free from slate and clinker-producing qualities. I have not heretofore secured *so high as eleven pounds* with small anthracite coals, but I am not sure that the coals used were of so good a quality as the ones to which these tests relate. The results obtained here are specially good, particularly those of October 3d and 6th, considering that the temperature of the escaping gases is some fifty degrees higher than usually accompanies the most economical work. The new condition of the boilers, and *their setting*, doubtless contributed something to these excellent results.

"The quality of the steam furnished was tested by means of a 'wire-drawing' calorimeter, attached to outlet 41, and the result was a continuous showing of dry steam. The indications of the thermometer varied from 288 degrees to 297.5 degrees, with a normal reading of 288 degrees at 85 pounds pressure, a continuous showing of dry steam. And at times there was an indication of three to four degrees of superheating."

DISCUSSION.

Mr. William H. Bryan.—I do not doubt that the boiler and setting described will give excellent results under favorable conditions. There are two points, however, which should not be overlooked:

First, the division wall in furnaces, and the "septum" wall at the rear end of boiler, will unquestionably reduce the capacity of the boiler for overwork, a consideration which cannot always be subordinated to high fuel economy and smokelessness, however desirable these features may be.

Second, the increased amount of fire brick work will do excellent service after it has once been thoroughly heated. To do this will require a considerable interval of time and a noticeable amount of heat. Where the boilers are in continuous service—as in water works, flour mills, etc.—this will not be objectionable, but where they are operated intermittently—say during the day only—there will be a loss of efficiency while the brickwork is being heated, and considerable heat will go to waste when it is cooling off after shutting down in the evening. In practice this has been found a serious objection to the use of increased fire-brick work.

I do not quite understand the author's statement that by passing the gases around his steam drum they are reduced to such a temperature as to render the use of a separate economizer unnecessary. In the evaporative trials reported in the paper the temperature of the escaping gases varied between 388 degrees and 450 degrees Fahr., averaging 423 degrees, which is probably as low as it is advisable to bring these gases with ordinary working steam pressures. It would certainly be possible—and often advisable—to reduce further the temperature of these gases 100 to 200 degrees by an economizer, with a corresponding increase in efficiency. Such a practice would be particularly desirable with condensing engines, where the feed water is only about 100 degrees Fahr.

It would be interesting to know the temperature of the hot air supplied under the grates, also whether there is really any advantage in its use. If its temperature resulted wholly—or in large part—from waste heat, the economy would not be questioned, as it must be when most of the heat comes directly from the furnace itself. Have any evaporative trials been made covering this point?

Mr. A. H. Eldredge.—Some years ago the writer was interested in a 62-inch x 16-feet boiler setting which, in some respects, was designed along the same lines as the one described by Mr. Le Van. The object sought for, as in his case, was the perfect burning of the fuel and its combustible gases.

It was a very ambitious setting, and had double walls with an 8-inch air space between them running the whole length and around the rear end of the boiler. The air under the grates was taken from openings through the front on each side of the boiler, and passed between the hollow walls next to the

fire box and then under the grates in a heated condition. The air used for the combustion of the gases entered the combustion chamber in the rear of the bridge wall through an air shaft, or tunnel, extending to the back end of the boiler and having openings from the tunnel to the combustion chamber through its entire length. The air was taken from the top of the boiler setting and travelled the length of the boiler three times before entering the air tunnel. The tunnel was built of fire brick, and was arched over on one side through an angle of 90 degrees. One object of the setting was to complete the combustion of the gases in the combustion chamber by adding heated air to the combustion chamber, rather than adding more air to the fire box, where the temperature is naturally very high, and, further, to take that air from the top of the boiler room, thus taking the first supply at a comparatively high temperature and also keeping a better and cooler atmosphere in the boiler room.

To utilize fully the products of combustion, they first passed under the shell of the boiler, then back as usual through the tubes, and then returned over the top of the boiler through a 24-inch chamber to the stack at the rear end of the boiler.

It might be well to note that one side of the boiler wall was built against a 17-inch building wall, also that there were peep holes through which the combustion of the gases could be observed through the length of the boiler. The draft plates in front of and on top of the boiler could be easily regulated at will. For the first three months after the boiler was in use the whole scheme which had been planned seemed to work to perfection. The products of combustion burned to a white heat underneath the whole boiler length; but, after that, trouble commenced, though no expense had been spared in either material or workmanship. The expansion and contraction of the walls soon caused the bricks in the arch of the air tunnels to work themselves down into the tunnel, until finally holes appeared which were not in the design of the setting. The side walls began to leak, and it was not long before the fireman was running the ash-pit doors wide open. The boiler was supported on iron columns entirely independent of the mason work.

In looking over Mr. Le Van's setting the writer naturally asks: Will not the bricks in arch 22, Fig. 28, gradually wedge

down and out of the arch? Again, the division wall, 30, is subjected to temperatures of 1,200 degrees or over. Should both these walls become weak, what is to prevent the boiler settling badly, possibly causing a serious accident? Again, it will be necessary also to support the boiler in some more or less difficult manner in case of repairs. Is not such a setting too expensive for common use? and would it not take a lock and key on the ash-pit doors to make the ordinary fireman use the proper drafts?

Now comes the question which has bothered the writer. One engineer supports a boiler on brickwork, and another, to avoid temperature effects, on a complicated system of levers. Which is right, or which, at least, is safe? Is there not some safe standard method that could be recommended to the public which can at least be applied to the old reliable fire-tube boiler?

Mr. Arthur C. Walworth.—This paper was of interest to me because I have had considerable to do with the construction and setting of horizontal tubular boilers, which have been of a very acceptable type in Boston, and have been used there a great many years, probably to a greater extent than any other type of boiler. I think there are some excellent points in this paper, but there are many others to which I should take exception. The first consideration is, that the expense of this Le Van boiler can be reduced without reducing the efficiency. If I were to construct a boiler of this horse-power, I would make it six inches larger diameter, say 60 inches, and have the steam space over the water, dispensing with the superheating drum. In Mr. Le Van's paper he shows a 54-inch boiler with a 16-inch neck to it, and a superheating drum over the boiler; the water line is carried high, using the drum as a separator. But if you take a 60-inch boiler at less cost, and the same amount of water surface in the tubes, you can carry the water line so far below the top of the boiler, that you do not need any superheating drum; and I doubt if the value of that superheater is equal to its cost. Of course where there is a series of boilers we have a connecting drum which has a proper connection to each boiler, and that acts as a separator in case any entrained water may go over to this drum. Of course when this boiler of Mr. Le Van's is set new, this superheating space is quite valuable, because it is perfectly clean. But as matter

of practice we find that spaces of that kind are not kept clean, and the top of the boiler soon becomes covered with soot and ashes, and eventually the whole surface of the superheater, unless it is cleaned every day. And it is my opinion that it would be much more economical to dispense with all this superheating arrangement, and take the products of combustion directly into a good economizer, reduce them lower than Mr. Le Van does in his experiment, and also dispense with this neck, which must weaken the shell of the boiler considerably. Of course with the old-fashioned dome on the boiler where we perforate the shell, the loss of strength in the shell is made up by the flanged rim around the steam dome. While twenty-five or thirty years ago we used to put domes on perhaps 25 per cent. of the boilers made, now a steam dome of that kind is an exception, because practically we find we get the required results with proper piping and connections, the same as with a steam dome. I do not think either that it is a good plan to take the feed water in through the blow-off pipe. Of course that would be a discussion by itself. The best practice in my mind is the simple one of taking the feed water into the steam space through a brass pipe, and letting it fall through the steam on to the water where it would not strike the shell anywhere.

Mr. Le Van's point about leaving a 4 or 5-inch space between the tubes and the boiler for the circulation of water, is a very essential point, and always ought to be observed, and it is also well to leave an extra space in the middle of the tubes for the downward circulation, the natural circulation of the water in the boiler of course being up from the hot plates on the sides, and that water has to be replaced by a current coming somewhere else; and the Boston boiler makers now usually make their boilers with a wider space between the central row of tubes, giving thus a circulation of water to every part of the boiler.

The boiler setting is quite interesting. I think there are a good many new points about this setting, and some of them have already been touched upon in the letters—such as the large amount of fire brick required and also the introduction of air at the bridge wall. I think Mr. Le Van has one good point in his boiler setting. He makes the wall next the boiler a 12-inch wall, and the outside wall 8 inches. I notice some of the "Hartford" boiler settings make the wall carrying the boiler 8 inches, and the outside one 12 inches. There is another question about

boiler setting on which boiler setters differ, and that is the necessity of binder bars on the outside of the boiler. I claim that if the boiler wall is properly set and properly built, with an inside 12-inch wall and an outside 8-inch wall, that the binder bars are simply superfluous. Those walls will stand just as well or better without them, but if the setting is cheap and badly built, you have to strap it together with iron bars and rods.

Relative to taking the air for the combustion under the grates through the walls of the boiler into the ash pit, I remember that Mr. Frederick Tudor of Boston, about fifteen years ago, had a plan of a boiler setting where he took the air in at various points from the boiler room, through a side wall and down in the air space, and then on the sides, under the grates into the ash pit. It did not heat the air very much, but in the case of a single boiler it was of more value than in a nest, because it kept both side walls cool and kept the boiler room cool by utilizing the heat that would otherwise be wasted. Practically it never was adopted to any great extent. The idea is not an especially new one. It has been tried at various times, and it is a good idea as far as it goes.

Referring to the hollow bridge wall, which we find discussed by Charles Wye Williams many years ago, we are using in Boston with considerable success, a device of Mr. Jones for introducing the air in that way, in which he brings the air in through the bridge wall through a series of brick tubes, or fire-clay tubes, in this way, the air being taken in from the ash pit, through, we will say, three doors in the front of the bridge wall and carried up into these tubes, or burners, entering the draft, in that way, on the back side. Of course, being in the flaming of the fire box, these tubes are hot, and the ash pit is hot, and the air that goes through these tubes issues so sufficiently heated as not to chill the products of combustion, but to immediately assist in combustion and make a proper flame in the combustion chamber at the rear of the bridge wall. Of course this is only one of many devices for bringing in sufficient air through the hollow bridge wall, but I mention it as one that has been quite successful.

There is a city ordinance in Boston requiring some smoke-consuming device to be used with all boilers which are used for bituminous coal, and this device has proved acceptable. Of course it is not an exclusive thing. The same end can be

attained in other ways. But I mention this as simply one way of accomplishing what Mr. Le Van accomplishes in another and much more expensive way in this boiler.

I would like to ask Mr. Le Van about the suspension of his boiler in the setting. I did not get at that exactly from the cuts, how he sets the boiler without lugs.

Mr. Rockwood.—Mr. Le Van is a courageous man, and perhaps that is why he endorses the shell and fire-tube boiler in the face of the "great progress" that has been made in boiler construction in the United States in the last ten years. I am glad some one has come out and said in a frank way that he likes the old fire-tube boiler best of all. I want to say that I do too.

Mr. F. W. Dean.—There may be no harm in this paper appearing in the *Transactions* of this Society, although I feel that it is something of a misfortune. I feel that it should receive sufficient discussion to prevent young engineers who consult papers for instruction from taking it too seriously.

The paper contains too many things to mention that are absurd, but when the author says that it is well known that superheated steam cannot exist in contact with water, he should be informed that it is better known that he is mistaken.

*Mr. Le Van.**—This boiler is one of the oldest type, and, although some seem to think it cannot be good on account of its antiquity, this fact does not justify some inferences drawn concerning it.

My forty years of experience with the old and the new types has not weaned me from my first love. Referring to the boiler trials at the Centennial Exhibition, you will find that the "Lowe" horizontal flue boiler stood at the head of the list. The water evaporated by it per pound of combustible at atmospheric pressure and temperature 212 degrees was 11.923; number of degrees superheated 8.885; whereas in the Babcock and Wilcox the water evaporated per pound of combustible at atmospheric pressure and temperature 212 degrees was 11.822; percentage of moisture in the steam, 3.24.

At the test of the boilers at the Brush Light Company's at Philadelphia, the horizontal flue boilers developed a horse-power with 9 square feet of heating surface, and the Babcock and Wilcox required 15 square feet to develop one horse-power. Now, if there is any improvement in modern pipe boilers of the hori-

* Author's closure, under the Rules.

zontal form over the old horizontal flue boilers, in what does it consist? I know a good thing when I see it, and when I find a good boiler I stand by it, if it *is* old.

Referring to the test made at Philadelphia by Mr. Barrus (see Table No 2, page 113), I would state that it was made under disadvantages, from the fact that there were only two boilers in place; that the chimney was of iron, 125 feet high and 6 feet internal diameter, and radiated off a great portion of the heat which required fuel to supply. The chimney has since then been lined with brick, reducing its diameter to 3.5 feet, which both reduced the coal consumption and increased the draft.

Mr. Bryan raises the objection to the construction and boiler setting under consideration, that the division wall of the furnace and the septum wall at the rear will reduce the efficiency of the boiler, but actual practice where I have introduced such features in the ordinary brick-set boilers proves the contrary.

Again, he says "there will be a loss of efficiency while the brickwork is being heated, and considerable heat will go to waste when it is cooling off after shutting down in the evening." "In practice this has been found a serious objection to the use of increased fire-brick work." This is not the fact. I have a plant of seven boilers erected in a cotton mill running eleven hours per day, the boiler pressure being 70 pounds per square inch. The fires are drawn every night, by seven o'clock, and a fresh fire started on a clean grate at four o'clock in the morning, the mill starting at six o'clock. The steam pressure during the night rarely falls below 60 pounds per square inch. The power developed through the engine by indicator diagrams shows 300 horse-power. The coal consumed averages 1,200 pounds per hour. This plant originally had eight boilers of the same heating surface set in the usual way and developing the same horse-power and consuming 1,400 pounds per hour. It was found after a few weeks' run that six of the boilers would perform the work, thus permitting one boiler to be out of service for cleaning. The steam is more regular and drier than before. With the old boilers the fires were cleaned about noon each day. The present boilers are run without cleaning during the eleven hours, thereby making less work for the fireman and also reducing the loss of fuel in the waste.

From the above it will be seen that the increased amount of fire brick does the same work that formerly required *eight* of

the old boilers, or, in other words, with thirteen per cent. less heating surface and twenty-one per cent. less grate surface, these boilers perform the same work, beside effecting a saving of one ton of coal per day. Further comment is unnecessary.

As to the use of a separate economizer, I would state that it is much cheaper and costs less to increase the diameter of the boiler six inches as involving less complication and cost of maintenance.

As regards the temperature of the hot air supplied under the grates, I am satisfied an advantage is obtained: the test made of the waste heat under the bridge wall exhibited in two cases a temperature of about 400 degrees.

Mr. A. Eldridge seems to think that the arch, 22, Fig. 28, will gradually wedge down and out of the arch. This cannot occur, as under the centre of the boiler is a fire-brick pier, 18 inches square, on which the rear end of the boiler rests. The arch, No. 22, is independent of this pier, and only carries the weight of the septum wall between hanging arch, No. 22, and boiler. The division wall between the fire chambers does not carry the weight of the boiler. The front end of the boiler has the bottom rim to project about 9 inches beyond the front head, one-half the periphery of the boiler to form the front flue (not shown in drawing); the bottom of this flue rests on a brick pier, No. 13, Fig. 24, reënforced by the fire front. This brick setting costs about $100 over the ordinary setting. And the saving of fuel will more than compensate for the additional cost.

Mr. Walworth's objection to the diameter of the boiler is all right, but when the specifications call for 54-inch diameter shell, it does not pay the contractor to put in a 60-inch diameter boiler. Mr. Walworth is mistaken when he says that you do not need any superheating drum. If he will carefully examine Table No. 2, Line 23, the stated evaporation per pound of combustible, it will speak for itself. In this test the calorimeter was applied to the rear pipe marked No. 41. "The indications of the thermometer varied from 288 to 297.5 degrees, with a normal reading of 288 degrees at 85 pounds pressure. There is continual evidence here of superheating, and this might be expected from the fact of the steam-heating surface, of which the steam issuing from this end of the drum had the benefit." *

* *Transactions* American Society Mechanical Engineers, vol. xi., p. 800.

Can Mr. Walworth produce any such results?

He argues that the superheating drum will soon become covered with soot, and should be dispensed with and an economizer substituted. Will not the economizer also become covered with soot? And are not these devices fitted with scrapers, run by power, to keep them clean? Which is the simpler and more economical, an engine to run the scrapers or the blowing of the superheating steam drum by a steam jet once a day?

In regard to the point made to the effect that the connecting neck weakens the shell, had he carefully examined the drawing (Fig. 22), he would see that both the neck connecting the superheating steam drum to the main shell of the boiler is reënforced by flanging outwardly the openings of the main shell, and that of the steam drum to fit inside the connecting neck is also flanged to fit the shells to which the neck is riveted.

He also objects to taking the feed water through the blow-off pipe, but fails to see that such pipe is of large diameter, and is encased in the hot walls of the boiler setting, and the water enters the boiler in a constant stream—not intermittingly—at a temperature of about the steam pressure. The suspension of the boiler without lugs is explained in my reply to Mr. Eldridge.

I agree with Mr. Walworth that if boilers are properly erected in brickwork, as stated by him, binder bars are superfluous. In my late practice I have dispensed with them.

A word in regard to Mr. Dean's statement that I am mistaken in my assertion that superheating cannot exist while the steam is in contact with water. My statement was made particularly in reference to horizontal boilers. I am aware that some vertical boilers, by reason of the contracted water space as well as the flue at the top, do superheat.

I have made some four hundred evaporative tests of horizontal boilers, and have never succeeded in getting superheated steam where water and steam space were combined.

Trowbridge, in *Heat as a Source of Power* (p. 35), says: "When heat is continuously applied to liquid in a closed vessel, the space enclosed remaining constant, the temperature of the liquid and vapor will continually increase if there be no dissipation of heat through the sides of the vessel; the quantity of the liquid will diminish and the pressure will increase, the vapor remaining saturated as long as any liquid remains. After the last element of liquid shall have passed to the state of vapor, if heat

be still applied, the vapor begins to assume the properties of the permanent gases. Under this condition, if the temperature could be indefinitely raised, or the volume of the space indefinitely increased, under a constant temperature, the vapor would finally arrive at a state corresponding to that of perfect gas, and would then follow strictly the law of Mariotte and Guy Lussac; and hence it may be said that the physical properties of vapors, when sufficiently expanded or at sufficiently high temperatures, and when they are not in contact with their liquids, are identical with those of perfect gases."

In my experience I have found that there is a considerable gain by superheating steam, particularly when the superheating is done by the gases of combustion after having passed the water-heating surfaces. That is my object in isolating the superheating space from the water-heating space by the use of a single neck connection at the extreme ends of each vessel, and the outlet for steam from the opposite end of the superheating drum, as time and space is an element for its completion (see Fig. 22—39, 40, and 41).

Mr. Dean has failed to furnish any such boilers to bear out his statement. No doubt he would have done so had he known of any. Mr. Barrus makes no such mention in the tests made by him in his large practice. He was present when the paper was under discussion, and had no remarks to make on the subject.

I beg to express my thanks to Mr. Rockwood for his kindly words relative to myself personally and to the subject matter of my paper.

I welcome support from such a capable source.

DCCXCVI.*

NOTE ON STRENGTH OF WHEEL RIMS.

BY ALBERT K. MANSFIELD, SALEM, OHIO.

(Member of the Society.)

On page 5 of Mr. Suplee's translation of *Reuleaux's Constructor* occur these words: "It is upon this point that the peculiar strengthening effect of ribs depends, and which makes their use so advantageous in cast-iron construction."

Then illustrations follow to support the fact that ribbed construction increases strength, that is, is economical of material; but it is not shown there, nor elsewhere to the present writer's knowledge, that this is not universally the case, but that the exact contrary may be the fact.

This may best be pointed out by a practical illustration:

The section modulus of the rectangular section of Fig. 30 is $\frac{bh^2}{6}$ or $\frac{26 \times 2\frac{1}{2}^2}{6} = 27.07$, AA being the neutral axis.

Now let one-fourth inch of the thickness of this section be formed into a rib $3\frac{1}{4}$ inches thick, as shown by Fig. 31. Preserving the same total area as before, this rib becomes 2 inches deep, making the total depth of the section $4\frac{1}{4}$ inches. The neutral axis will be found to be located as shown in the figure, and from a well-known formula the section modulus is found to be 18.64; that is, the least section modulus, or the moment of inertia divided by the greatest distance of fibre from the neutral axis.

The section has therefore been weakened by the ribs; in fact, unless it represents the special case of a cast-iron beam loaded so that the extreme fibre is subjected to compression, it is only about two-thirds as strong as before. It is seen, therefore, that under some conditions ribbing does not have a strengthening effect. It would clearly be useful to generalize this matter in

* Presented at the New York meeting (November, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

such a way as to make it easy for the designer to determine when and to what extent ribbing may or may not be resorted to with advantage, but that is beyond the purpose of the present writing.

If we separate the rib of Fig. 31 into two ribs and place them at the ends of the section as in Fig. 32, we have, without change of strength, a section similar to that often used in the construction of wheel rims. The bending moment of such rims, due to centrifugal force, acts with maximum effect near the wheel arms, in such way as to cause *tension* at the extreme fibre. It will be clear, therefore, that such ribs in such place may make the wheel weaker than it would be if they were not employed. It is not meant that wheels cannot be ribbed in this manner to their advantage, but rather that it is necessary to take care that they are not thus ribbed to their disadvantage. There are many wheels in service which have been strengthened in this negative way.

The destructive accidents which have occurred to large wheels have led to bringing cast-iron into disrepute as a material from which to construct such wheels. This fact, coupled with that pointed out above relative to rim flanges, has led the present writer to the use of the method of ribbing wheels illustrated in Fig. 33, which seems to fill all requirements, enabling cast-iron rims to be made as strong as may be practically desired.

Maintaining the same face of wheel and total area of section as before, let us employ $\frac{3}{4}$ inch of the thickness of rim in forming a rib $1\frac{3}{4}$ inches thick at centre of rim; the section then becomes as shown by Fig. 34. The neutral axis of this section is at line *AA*, and the section modulus becomes 77.5, which is nearly three times as strong as the section of Fig. 30, and more than four times as strong as the flanged section of Fig. 32. Of course, a further increase of strength may be obtained by adding to depth of flange.

Now the section of Fig. 34 is taken to represent that next the arms, in which (the case being that of a built-in beam uniformly loaded) the extreme fibre is under tension.

The mid-section between arms, however, while acted upon by an equal moment (when the cross section is uniform), is bent in an opposite direction; therefore the extreme fibre is compressed, which, for equal strength, enables the flange at this point to be reduced, because cast iron is twice as able to resist compres-

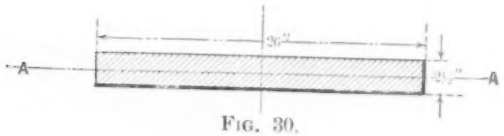


FIG. 30.

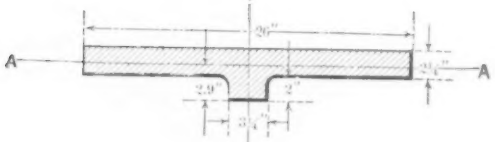


FIG. 31.

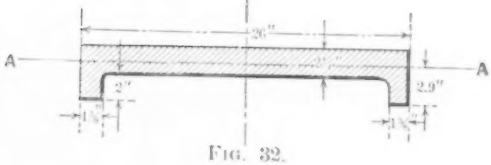


FIG. 32.

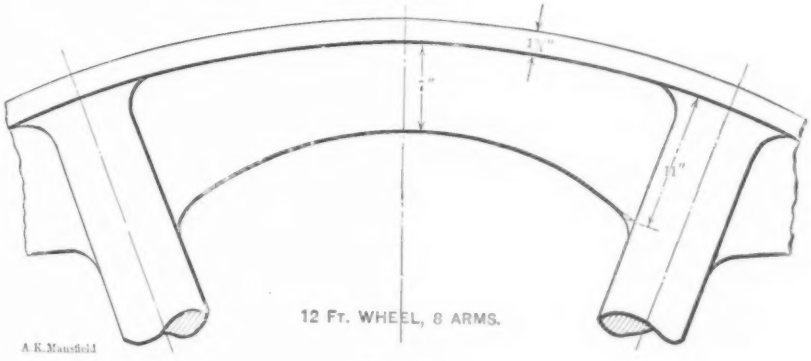


FIG. 33.

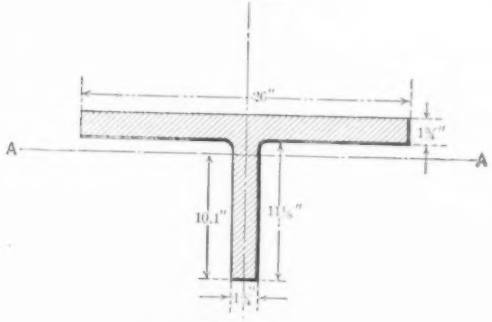


FIG. 34.

sion as extension. The section modulus at this point may therefore be one-half as great as at the arms, which will be found to correspond with a flange depth of about five-eighths of that at arms, or in this case about 7 inches.

These proportions are shown in Fig. 33. Almost any desired strength may be attained by this method. It has several other advantages over uniform flanges at edges which will readily occur to designers of wheels.

DISCUSSION.

Prof. D. S. Jacobus.—Professor Denton, who is unavoidably absent this morning, has asked me to bring up the point that by casting a flange on the rim in the way represented in the author's second sketch, we do reduce the strength of the rim, but we increase the stiffness, and the stiffness is an element of strength. Ordinarily speaking, when a fly-wheel bursts there is some irregular action, and it appears that the irregular action causes a pulsation in the wheel, and that it is this pulsation which augments the strain and breaks the wheel. By increasing the stiffness we tend to decrease the pulsations, and thus make the wheel stronger, and this fact has not been considered by Mr. Mansfield.

Mr. William S. Aldrich.—Mr. Mansfield has very cleverly shown us how the process of evolution of a ribbed wheel rim may be carried through a series of faulty proportions to one of recognized value. It is extremely important to notice that he is dealing with the same amount of material. It will cost no more to make the wheel rim much stronger by properly disposing the material. A recognition of this feature of the economics of machine design might be profitably extended to many other details. The effect of cooling on the ideal rim section, as shown in Fig. 34, will be very marked compared to that on the prevalent type, shown in Fig. 31. Making flange and web of the same thickness, or very nearly so, provides for the ideal cooling of cast iron or of cast steel. Initial strains are avoided which, if we mistake not, have sometimes caused fly-wheels to burst while lying in the foundry yard; and to unequal cooling may be traced many an accident after the wheel has been put into service. Cast metal will not be so treacherous if treated in this rational manner, as suggested by Mr. Mansfield. It is apparent

that some ribbing should be provided to resist the so-called bursting pressure, due to centrifugal force; but the new outline of this rib, as shown in Fig. 33, is as novel as it is interesting, as an example of the application of the principles of structural mechanics to fly-wheel designing. It is clear, also, in following out this application, that the segments of fly-wheels should be bolted together midway between the arms, and not at the arms, as is often done.

Mr. H. H. Suplee.—As Mr. Mansfield refers to my translation of *Reuleaux's Constructor*, I think that in justice to him the quotation should be carried a little further. When the entire paragraph is quoted, I think the subject is placed in rather a different light. Professor Reuleaux says, after referring to the question of strengthening by flanges or ribs:

* "It is a matter of some importance for the designer to keep in mind some general conclusions which may be drawn from the tables as to the influence of various shapes upon the strength. It will be plainly seen that the depth of a section is the dimension which has the greatest influence upon the strength, and also that those portions of the section which are furthest removed from the neutral axis are of the most service.

"It is upon this point that the peculiar strengthening effect of ribs depends, and which makes their use so advantageous in cast-iron construction. These ribs do not act so much by the mere strength of their own cross section as by the fact that they strengthen those portions which are furthest removed from the neutral axis."

Then follows a practical example of almost identical nature with that given by Mr. Mansfield.

I desire to put this on record so as to present Professor Reuleaux in the proper light.

Mr. Teile H. Müller.—In the October (1893) number of the always valued *Power*, I found some notes on a fly-wheel accident in Memphis, which, I think, admit of some additional remarks.

Although the data given in the account of the accident were very meagre, it appears that there are a sufficient number of dimensions, etc., to arrive at some conclusions regarding the wheel, and from these it would appear that the strains produced

* *Reuleaux's Constructor*, translated by H. H. Suplee, page 5.

may come very near to ultimate strength, and exceed the elastic limit to a considerable extent at the normal speed of 152 revolutions per minute.

In a wheel with a deep narrow rim we are no doubt justified to calculate the centrifugal effect for the entire rim, assume this to be an evenly distributed load acting radially outward, and take the projection of all these forces on the diameter as the strain produced in any diametrical section of the wheel, in the same manner as we calculate the strains produced by internal pressure in a tube, and so far I agree perfectly with the results as reached by others.

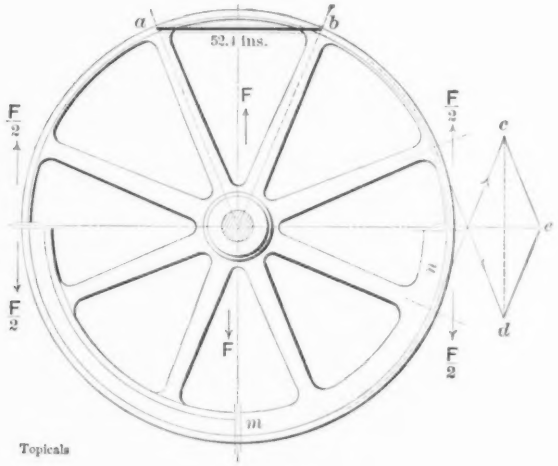


FIG. 35.

When, however, the rim is wide and thin, as the rims of pulleys are, and the distance between the arms where they are joined to the rim large, when compared to the thickness of the latter, we shall have to examine that part of the rim extending from one arm to the other for other strains than those found by above general formula for centrifugal force (Fig. 35): $F = \frac{w.\ v^2}{gr}$.

If we had a perfectly flexible rim, say one made of thin sheet steel, and permitted the part $a'b$ to assume the most favorable curve due to the different forces, it would not be very difficult to find the tangential forces as well as the shearing strains at a and b. But we would not have a round wheel.

For cast iron, however, and for the small deflections permissible to keep the rim to its true circle, we shall be compelled to regard the piece from a to b as a rigid beam, and calculate the strains appearing under that assumption.

Judging from the illustrations in *Power* for October 1, 1893, I assumé that the wheel had two sets of arms, eight in each set, and from data given—

Diameter of wheel	11 feet 6 inches = 138 inches.
Face of wheel	50 inches.
Thickness of rim about	1 inch.
Revolutions per minute	152 inches.

This would give a mean diameter for centre of rim of 137 inches, and $r = 5.7083$ feet, and v (velocity in feet per second) $= 91$ feet—$v^2 = 8281$—when the wheel is remaining at a normal speed of 152 revolutions per minute. At this speed we find from the above formula that every pound of metal in the rim produces a centrifugal effect of

$$F = \frac{w\,v^2}{g\,r} = \frac{1 \times 91^2}{32.16 \times 5.708} = 45 \text{ lbs.}$$

A part of the rim extending from a to b, being 50 inches wide, and having two inwardly projecting edge rims (the dimensions of which are not given), would weigh nearly 800 pounds, and therefore $800 \times 45 = 36{,}000$ pounds, would represent the uniformly distributed load to be sustained on this part of the rim.

We shall not be very far from the truth if we consider the section of rim from a to b (taking into account the nature of cast iron and the rigidity required) as a beam uniformly loaded and firmly fixed at the ends, and at the same time, to simplify figures, take only a strip one inch wide for calculation.

We found above the centrifugal effect for normal speed of wheel 45 pounds for each pound weight, and taking cast iron at 0.26 pound per 1 cubic inch, we find that each inch in length of our beam 1 inch wide extending from a to b has to sustain 11.7 pounds. Projecting this load on to the chord ab, we find that we have a total load of 613 pounds on that beam uniformly distributed.

The strength of a beam firmly fixed at each end, and uniformly distributed load, is found by the general formula.

When P = uniformly distributed load.

$P \times L = 12\ R \times S$.

L = Length of beam in inches.

R = Moment of resistance of the cross section.

S = Strain per square inch in most strained fibre in pounds. Transposing this formula for fibre strain, we have

$$S = \frac{P.L}{12.R}.$$

In our case we have L = length of beam = 52 inches.

R = moment of resistance for rectangular section = $\frac{bh^2}{6}$, and for a beam 1 inch wide and 1 inch thick (the rim being 1 inch thick) $\frac{bh^2}{6} = \frac{1 \times 1^2}{6} = \frac{1}{6}$.

For $P = 613$ pounds, S becomes $\frac{613 \times 52}{12 \times \frac{1}{6}} = 15{,}938$ pounds.

To this fibre strain will have to be added the strain resulting from the centrifugal forces acting on the wheel, and tending to separate it in two halves, as found by the formula cited for a hollow cylinder, which in the present case amounts to some 800 pounds per square inch, designated in Fig. 35 by $\frac{F}{2}$. Owing to the inwardly pointing flange on the rim, the moment of resistance will be somewhat larger than that of a plain rectangle, but not of sufficient influence to affect the result very much. The above figures show that the section of the rim for a given weight must show a different distribution of metal, either with a rib between the arms as in Fig. 36 (and also in Fig. 35 from m to n), or, better yet, as in Fig. 37, which is a variation of the Fairbairn section of cast-iron girder, Fig. 38, with flanges proportioned to the capacity of cast iron to resist compression and tension.

Is not the high strain found above corroborated by the description of the accident by the engineer, as quoted, "that the wheel seemed to open up altogether, without warning?"

The proportioning and shaping of a proper section for the rim will not present any serious difficulty to either the designer or the foundryman, whose requirements should be carefully considered in such work.

I trust that the above sketch may prove to be of some value

to those interested in this fly-wheel rim question, and perhaps induce some gentlemen having sufficient time to evolve complete formulæ for their fly-wheel rims at high speed, which take notice of the shearing and bending strains appearing at given speeds. But even then in making such a wheel we shall have

FIG. 36.

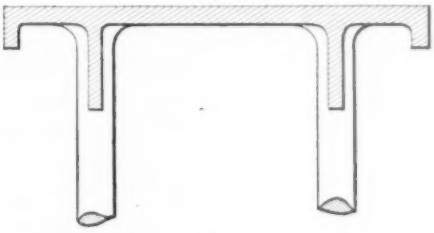


FIG. 37.

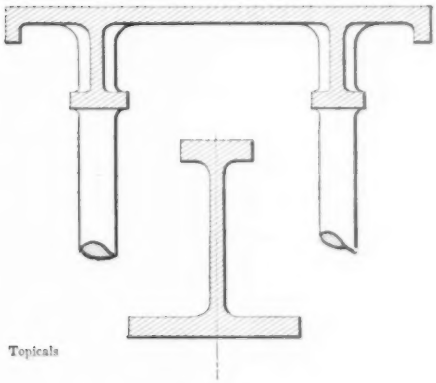


FIG. 38.

to consider quite a number of points that cannot be expressed in dx/dy.

The point I wish to make is, that strains other than the plain and simple tangential strains resulting from centrifugal forces, exist in the thin rims of cast-iron fly-wheels, which strains may be serious and even dangerous at the normal speed for which the wheel is intended. On the other hand, in this apparent epidemic of hurting fly-wheels, papers can do and are doing an enormous amount of good, not only by the information imparted through their editorials, but by the liberality shown

in opening their columns for the general discussion of the subject.

Prof. J. Burkitt Webb.—If I understand the discussion, it is assumed that the rim of a fly-wheel running at a high speed is bulged out between the arms by centrifugal force, and Professor Unwin is referred to as assuming that it does. Now, so far as any book work is concerned, an author usually takes care to simplify a problem to suit his mathematics, and I do not think that any one has allowed for the stretch of the arms without considering them to be of uniform section. The question which occurs to me is whether it might not be possible to so proportion the arms that the wheel would remain round under the influence of centrifugal force, and I am quite sure that an arm could be so proportioned, and even that it might be proportioned so that the arms would stretch out farther than the rim. This becomes evident by considering that an arm of uniform section stretches about one-sixth of the right amount, and that the other five-sixths could be obtained by swelling out the outer end of the arm into a mass about equal to that of the rest of the arm.

But this must not be construed as a proposal to build fly-wheels in this way. It seems to me that in the construction of fly-wheels of large size a lesson might be learned from bridge construction, in which no one ever thinks of using cast iron for tension members. Why not cast the rim of the wheel in any suitable lengths and join them so as to receive compression, not tension. Then make the web of numerous steel rods, bicycle-fashion, arranged tangentially to the hub, of course. I believe that such a construction would be better and cheaper than any other, for it must be noticed that the stress on all the parts could be easily and certainly calculated, and therefore no waste metal or large factor of safety would enter into it. In such a wheel enough initial tension would be put into the spokes to prevent there ever being any tension in the rim.

*A. K. Mansfield.**—Professor Denton's point that shallow flanges increase stiffness though they may reduce strength is correct, but he probably does not intend to offer this as a reason why we may sacrifice strength, which is an important quality in wheels, and in order to avoid overstraining the metal, must be considered for itself.

* Author's closure, under the Rules.

Professor Aldrich points out that with the ribbed construction of Fig. 33 it is logical to make the joint of built-up wheels midway between arms, and treat the rim as a series of cantilever bridges. Certainly if due strength is provided for, this is as rational as is the cantilever bridge, and the method of ribbing suggested by Fig. 33 is adapted to meet all practical needs as regards strength.

Mr. Suplee's more extended quotation from *Reuleaux's Constructor* does not appear to alter the fact that this author, in common with authors of other text books, fails to point out that flanges or ribs *may weaken a beam*. He treats of the subject of "strengthening by ribs," but not of that of weakening by the same means. He refers to the "advantageous use of ribs in cast-iron construction," but nowhere to the fact of their possible disadvantageous use.

In this respect he does differ from other writers, but expresses the seemingly general view in a conveniently quotable way.

I am sorry that this matter has not received more attention from the present discussers, not in its relation to wheel rims alone, but in the general subject, which appears to have novelty.

DCCXCVII.*

THE VARIATION OF BELT TENSIONS WITH POWER TRANSMITTED.

BY WM. S. ALDRICH, MORGANTOWN, W. VA.

(Member of the Society.)

THERE are certain questions continually arising in engineering practice which do not seem to admit of settlement by either reason or experiment. Some of these ever-recurring questions relate to the transmission of power by belting. In what way are the belt tensions altered as the load is applied? What effect has the change of load on the sum of the belt tensions? Is there any relation between the belt tensions which does not involve the coefficient of friction?

The present paper does not attempt to settle these or similar points by reporting new experiments on belting. Its purpose is to open up a discussion of such mooted questions, perhaps in a different line from that heretofore considered. From a careful study of the "Experiments on the Transmission of Power by Belting," made by Messrs. Wilfred Lewis and J. S. Bancroft for Wm. Sellers & Co., of Philadelphia, and reported to this Society,† the author is led to the following conclusions:

That, with a given initial or laced tension and a constant lineal speed, when the load is increased:

(1) The tension on the tight side of the belt increases much more rapidly than the tension on the slack side decreases.

(2) The working tension $(T_1 - T_2)$ increases more rapidly than the sum $(T_1 + T_2)$ of the tensions on the tight and slack sides.

(3) The working tension $(T_1 - T_2)$ tends continually to approach the value of the sum $(T_1 + T_2)$ of the tensions, as the load is increased indefinitely as far as the breaking strength of the belt.

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* Presented at the New York meeting (December, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

† *Transactions* A. S. M. E., vol. vii., No. 213, "Experiments on the Transmission of Power by Belting," by Wilfred Lewis.

(4) The ratio of the tensions of a belt transmitting power cannot be calculated with any degree of accuracy by means of the well-known belt formula $\left(\frac{T_1}{T_2} = e^{f\theta}\right)$ involving the arc of contact and the coefficient of friction.

(5) There is a definite relation between the ratio of the belt tensions and the ratio of their sum and difference, which relation is independent of the coefficient of friction, and may be reduced to the following form:

$$\frac{T_1}{T_1 - T_2} = e^{\frac{2T_2}{2T_1 - T_2}}.$$

Increasing the power transmitted by a belt having a definite arc of contact on the pulley and moving at a constant lineal speed, has a twofold effect: it produces an increased slipping of the belt over the pulley, and results in a re-distribution of the forces in the belt itself. These alter the respective values of the tensions on the tight and slack sides, increasing their ratio, increasing their difference in proportion to the power transmitted; increasing, also, their sum, and so increasing the pressure on the journal. In the midst of all of these changes, the coefficient of friction undergoes a radical alteration in value, which will be considered in the discussion of the well-known belt formula, involving the coefficient of friction as a function of the ratio of the belt tensions.

The difference of the belt tensions is a direct measure of the power transmitted, and is therefore the most readily obtained value in any case of belt transmission. It is sometimes called the working tension, or the effective tension, or the driving tension. It steadily increases from a zero value, when the belt is at rest, to a value approaching more and more the sum of the belt tensions as the load is increased.

The sum of the belt tensions has the least value when the belt is at rest, and is then equal to the initial or laced tension. It has now been incontestably proven that the sum of the belt tensions does not remain constant as the load is increased. Mr. Lewis sums up the matter thus: "that the sum of the belt tensions remains constant, is directly controverted by these experiments, and it can now be positively asserted that in all cases an increase in the load is accompanied by an increase in the sum

of the tensions." This results in an increase of pressure on the journal, and may be so measured in any given case admitting of such measurement, notably so in the case of dynamos and motors mounted on sliding ways.

The writer has used the terms "tight" and "slack" sides of the belt, rather than the older terms of "driving" and "following" sides, for the reason that the slack side is just as much a factor as the tight side in transmitting power by the belt. The slack side of the belt, while it does seem to follow the tight side in point of location, has a tension that is an all-important element in determining the effective or working tension to which

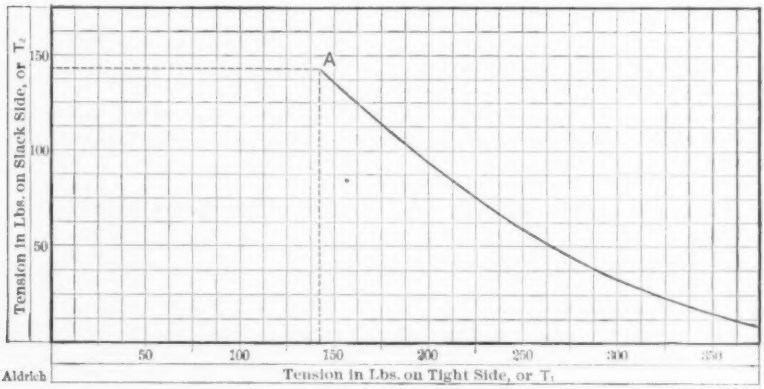


FIG. 39.—SHOWING THE VARIATION OF THE BELT TENSIONS, T_1 AND T_2, AS THE LOAD IS INCREASED. AT THE POINT "A" THE TENSIONS ARE EQUAL AND THE BELT IS AT REST.

the belt is subjected in any given case. In other words, the slack tension is an element in the driving tension of the belt.

The many valuable experiments recorded in Mr. Lewis's paper, previously referred to, furnish sufficient data for a few generalizations. Two of these of the first importance relate to the variations in the individual belt tensions as the load is altered, and to the manner in which the sum and difference of the tensions vary under like conditions. Characteristic curves for each of these relations are plotted in Figs. 39 and 40. It is quite probable that a careful comparison with similar mathematical curves will show to what particular type these belong; or their equations may be derived by the well-known method of obtaining empirical formulæ to represent the relations given by the

curves plotted from observations. The author has not yet found any satisfactory relation between the quantities involved which will apply to Mr. Lewis's experiments in general. It may be interesting to note that Fig. 40 represents the variation of the pressure on the journal with the horse-power transmitted,

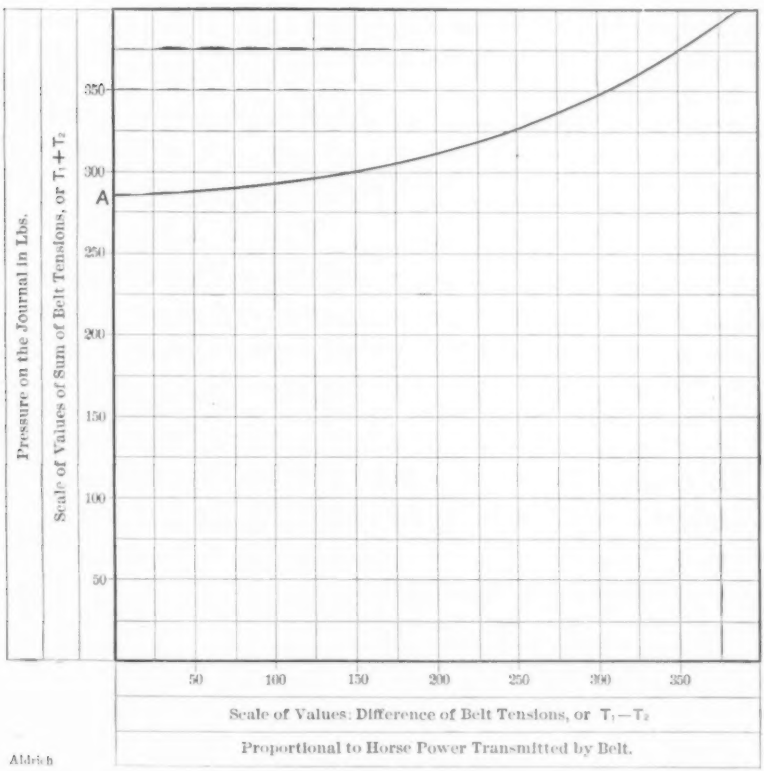


FIG. 40.—SHOWING THE VARIATION OF THE SUM AND THE DIFFERENCE OF THE BELT TENSIONS, OR THE INCREASE OF PRESSURE ON THE JOURNAL AS THE HORSE-POWER TRANSMITTED BY THE BELT IS INCREASED. AT THE POINT "A", THE PRESSURE IS DUE TO THE INITIAL OR LACED TENSION AND THE BELT IS AT REST.

since these are directly proportional to the sum and difference of the belt tensions, respectively. When a satisfactory relation is found connecting these two quantities we shall be able to calculate at once the pressure on the journal due to any horse-power transmitted by the belt. The importance of this, in some cases, cannot be overestimated.

Another result of this determination will be a more accurate way of obtaining the average strain to which the belt has been subjected under any given load conditions. Mr. F. W. Taylor, in Sec. 75 of his "Notes on Belting," read before this Society,* calls attention to the unsatisfactory state of our present knowledge on this point:

"It would seem as though the effective pull of the belt on the rim of the pulley might be calculated by considering the tension of the cone belts at the time when they required tightening in connection with their coefficient of friction and the well-known belt formula. However, in looking through the careful experiments to determine the coefficient of friction, we see that the coefficient of friction varies from 15 per cent. to 135 per cent., according to the nature and condition of pulleys and belts, the belt dressing, the amount of slip, and even the atmospheric conditions. And, again, the notable discovery of Messrs. Lewis and Bancroft (whose experiments, on the whole, appear to be the most carefully and thoroughly conducted), that the 'sum of the tensions on both sides of the belt does not remain constant,' upsets all previous theoretical belt formulæ. When we consider these facts it would seem doubtful whether any calculation on the subject would lead to reliable results."

The old established formula: $\frac{T_1}{T_2} = e^{f\theta}$

is based on assumptions that do not hold true in the transmission of power by belts. It is derived on the assumption that the belt is just on the point of slipping, but that slipping does not actually occur. In point of fact, slipping does occur, and to a remarkable extent. It continues to increase as the load is increased, effecting radical and almost indeterminate changes in the coefficient of friction. Many of the experiments of Mr. Lewis show that with steady load and at constant lineal speed the belt kept on slipping, in some cases till it slipped off. Under these conditions the ratio of belt tensions remained constant because of the steady load on the belt; and yet this would imply, by the application of the theoretical formula, that the

* *Transactions* of the A. S. M. E., vol. xv., No. 568, "Notes on Belting," by Fred. W. Taylor.

coefficient of friction remained constant. While it may be true, as earlier pointed out by Mr. Towne, "that the speed of slip between the belt and the pulley is a factor of prime importance in determining the coefficient of friction," clearly "there is still need of more light on the subject," as pointed out by Mr. Lewis at the conclusion of the whole summary of his experiments on belting.

Fig. 41 shows the variation of the ratio of the belt tensions with the coefficient of friction, as determined from this ratio by the theoretical formula. It has not been possible to obtain any

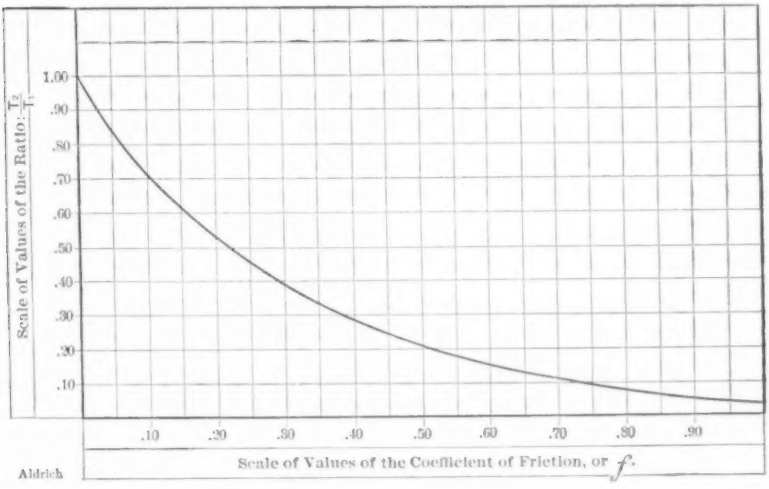


FIG. 41.—SHOWING THE VARIATION OF THE RATIO OF THE BELT TENSIONS WITH THE COEFFICIENT OF FRICTION.

relation between the speed of slip and the coefficient of friction, even by an added study of Professor Lanza's valuable experiments, as reported to this Society* in his paper on the "Transmission of Power by Belting," whose closing lines breathe a similar spirit of hopefulness for the future: "While there are doubtless some discrepancies in these few experiments which will be eliminated as soon as we have a larger number, and while we shall before long be able to make out a table showing exactly how much we must strain our belts to accomplish any desired

* *Transactions* A. S. M. E., vol. vii., No. 202, "Transmission of Power by Belting," by G. Lanza.

result as to transmission and slip, what has been done is sufficient to enable us to say with certainty that a coefficient of friction of 0.42 is altogether too large and is never realized in practice."

Upon plotting a series of values from Mr. Lewis's experiments to show the relation between the ratio of the tensions and the ratio of their sum and difference the general curve, Fig. 42, was obtained.

In order to put the curve in another form closely allied to

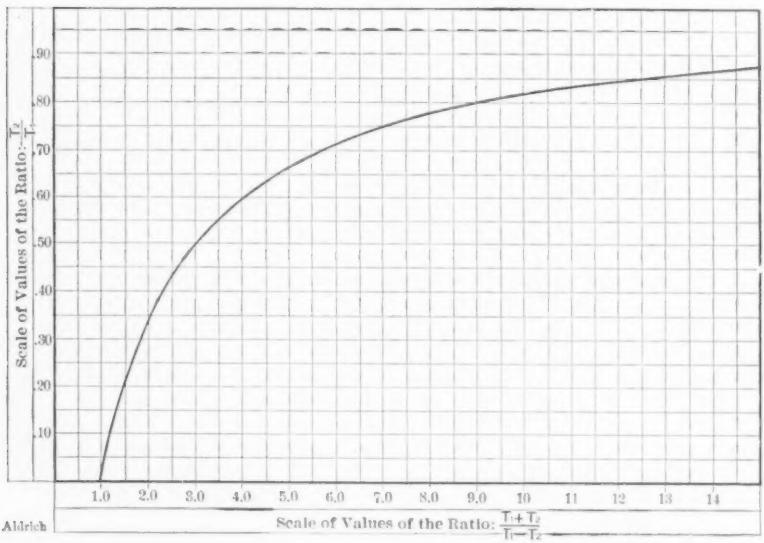


FIG. 42.—SHOWING THE VARIATION OF THE RATIO OF THE BELT TENSIONS WITH THE RATIO OF THEIR SUM AND DIFFERENCE.

an existing mathematical curve, Fig. 43 was plotted showing the relation between the value of unity minus the ratio of the tensions, and the value of the ratio of the sum and difference of the tensions minus unity. This is clearly a curve of the logarithmic form, and upon applying the values from the experiments to such a curve, they were shown to agree in a remarkable manner. From the equation of the logarithmic curve of this type:

$$y = e^{-\frac{x}{k}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

we may, therefore, write the following equation for the curve of Fig. 43:

$$1 - \frac{\tau_2}{\tau_1} = e^{-\frac{1}{k}\left(\frac{\tau_1 + \tau_2}{\tau_1 - \tau_2} - 1\right)}. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The value of the constant (k) was found by repeatedly applying the equation to the results of the experiments reported by Mr. Lewis, and found to be of the form:

$$k = 1 + \frac{\tau_1}{\tau_1 - \tau_2} = \frac{2\,\tau_1 - \tau_2}{\tau_1 - \tau_2}. \quad . \quad . \quad . \quad . \quad (3)$$

Substituting this value, we have:

$$\frac{\tau_1 - \tau_2}{\tau_1} = e^{-\frac{2\,\tau_2}{2\,\tau_1 - \tau_2}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

or, written in another form, as follows:

$$\frac{\tau_1}{\tau_1 - \tau_2} = e^{\frac{2\,\tau_2}{2\,\tau_1 - \tau_2}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The value of e, the base of the Naperian system of logarithms, is 2.7183.

This equation has been equally tested by applying it to the recorded data of the two independent investigators, Mr. Lewis and Professor Lanza, for conditions throughout a wide range. From one with a coefficient of friction of 0.228 and a belt slip of 0.8 foot per minute to one with a coefficient of friction of 1.37 and a belt slip of 54.8 feet per minute, the differences between the actual and the calculated values (as given by the formula) pass from very small positive to very small negative values. From these and like considerations it would appear that the disturbing elements of belt slip and variable coefficient of friction do not enter to change in any very material way the relation between the ratio of the belt tensions and the ratio of their sum and difference, as finally reduced to the form of the equation referred to.

The formula derived from a consideration of the logarithmic or exponential type of curves, as shown in Figs. 42 and 43, is clearly an empirical one. It is not claimed that it will hold true throughout all ranges of loads for all cases of belt transmission. Such a formula is yet to be derived. But the present one approximates so closely the results of the experiments of Mr. Lewis and Professor Lanza, two independent investigators, that

it may be taken for all of the ordinary cases of belt transmission. It is at least one form of expression for the general law, so clearly indicated by Figs. 42 and 43. The problem is to arrive at an expression for the ratio of the tangential and normal forces on the belt as it moves over the pulley in transmitting power. In other words, what is wanted is the coefficient of friction of the belt on the pulley under dynamic conditions, when transmitting power, and not the coefficient of friction under statical conditions, when just on the point of slipping. What such a

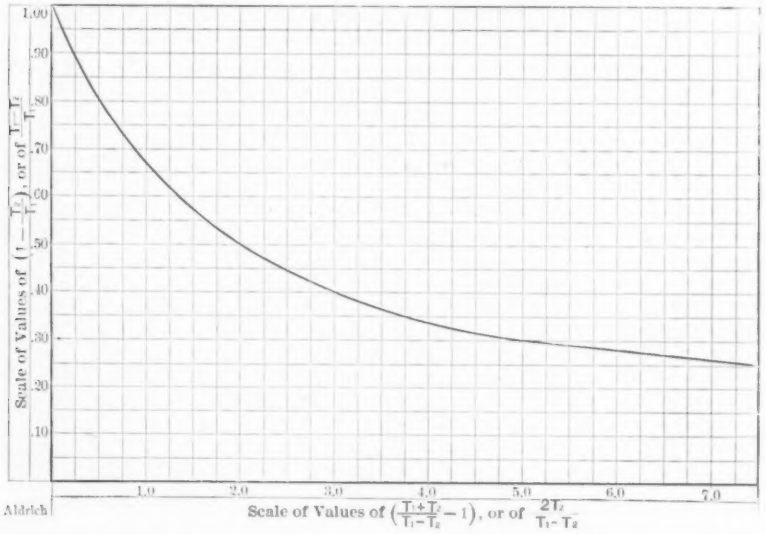


FIG. 43.—SHOWING THE SAME LAW OF VARIATION AS FIG. 42, AND FROM WHICH THE EMPIRICAL FORMULA WAS DERIVED.

relation should be has not yet been determined; but it does seem as though it should involve the tangential or effective pull of the belt (the difference of the belt tensions, $T_1 - T_2$) and the pressure on the journal (the sum of the belt tensions, $T_1 + T_2$). The effect of centrifugal force on the belt should be considered theoretically; but this is taken into consideration in the experimental data themselves, which give the values of the sum and difference of the belt tensions. The many unsteady and uncertain conditions of belt testing make it difficult to get exact relations, much less to duplicate them. We shall await the results of other investigators to corroborate these conclusions drawn from the experiments heretofore presented to this Society.

DISCUSSION.

Mr. Carl G. Barth.—Though the experiments on the transmission of power by belting made by Messrs. Lewis and Bancroft date back to a time when I was still in the employ of William Sellers & Co., of Philadelphia, and though I was at one time fully familiar with Mr. Lewis's paper on the subject, I must confess that I have never seriously paid any attention to the conclusions drawn by Mr. Lewis until reading over the paper now presented to this Society by Mr. Aldrich. In so doing, however, the idea suggested itself to me, that all this mystery about the behavior of the tensions in the two sides of a belt, could probably be explained on the quite plausible assumption that the modulus of elasticity of leather is not practically constant for all stresses within the elastic limit—as it is for most materials of construction—but decidedly variable. Acting on this suggestion, I readily found that a curve of the same general character as that plotted in Fig. 39 of Mr. Aldrich's paper, to represent observed simultaneous tensions in a belt, could be constructed on the assumption that the elongation of a piece of leather is proportional to the square root of the stress, instead of to the stress direct; and encouraged by this, I soon reproduced the curve itself with remarkable closeness, on the assumption that the elongation varies as the .65 power of the stress. Assuming thus:

$$\frac{\Delta l}{l} = \frac{T^{.65}}{C},$$

where Δl is the elongation of a length l under a tension T, and C is a constant, we may write the excess in the length of the light side of a belt over its original length l, under no tension,

$$\Delta l_1 = \frac{T_1^{.65} l_1}{C}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and, similarly, the excess in length of the slack side,

$$\Delta l_2 = \frac{T_2^{.65} l_2}{C} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Adding (1) and (2) gives

$$\Delta l_1 + \Delta l_2 = \frac{T_1^{.65} l_1 + T_2^{.65} l_2}{C} \quad . \quad . \quad . \quad . \quad (3)$$

But when no power is transmitted, T_1 and T_2 are equal, and designating their common value by T_m, we may therefore write

$$2\,\Delta\,l_m = 2\,\frac{T_m^{.65} l_m}{C} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Now, it is clear that the total elongation of a belt must be practically the same, no matter how the tensions vary, and no matter how influenced by centrifugal action, so long as the belt remains in contact with its pulleys, and hence we may write

$$\Delta\,l_1 + \Delta\,l_2 = 2\,\Delta\,l_m,$$

so that combining (3) and (4) we get

$$T_1^{.65} l_1 + T_2^{.65} l_2 = 2\,T_m^{.65} l_m.$$

But no appreciable error will be committed by writing l_m for both l_1 and l_2, which, however, will simplify this last equation to

$$T_1^{.65} + T_2^{.65} = 2\,T_m^{.65}. \quad . \quad . \quad . \quad . \quad . \quad (5)$$

On the above made assumption, and neglecting the undoubtedly only small possible modifications that would be introduced by taking account of the variable tensions and elongations in the parts of a belt in contact with the pulleys, and the variation in the catenaries of the open parts, we thus have a very simple equation for the relations existing between the tensions in the two sides of a belt-transmitting power, and the initial tension of the belt at rest; and the same is hereby respectfully submitted to the consideration of such investigators as have a good testing machine at their disposal, and who are thus able to determine whether the assumption made will be borne out by the facts about the physical properties of leather. However, even if not substantiated by such facts, this formula would, on account of its greater simplicity, be a material improvement as an empirical substitute for the one that constitutes Mr. Aldrich's fifth conclusion. In either case it implies the acceptance of Mr. Aldrich's first three conclusions, while I fail to see that he has in any way substantiated his fourth conclusion—the condemnation of the *well-known belt formula* $\left(\frac{T_1}{T_2} = e^{f\theta}\right)$. I thus do not see why the friction of a slipping belt should be essentially different from that of a *statical* one, except in so far as it has been well established that the coefficient of

friction for slip is greater than for rest, and thus explains why a belt must slip considerably in order to transmit an amount of power that calls for a great difference in the two tensions, and accordingly also a great ratio between them. To be sure, the formula is in so far defective as in its derivation the effect of the thickness of the belt is neglected, this being treated as infinitely small as compared with the diameter of the pulley; and I therefore take this opportunity to also offer an improvement on this formula, which I derived some time last spring while working up the subject of power transmission for a new advanced course of mechanical engineering for the International Correspondence Schools of Scranton, Pa.

This improved formula is

$$\frac{T_1}{T_2} = e^{f\theta \frac{d}{\sqrt{(d+t)^2 + (2dt + t^2)f^2}}}$$

which without material error may be written in the simpler form,

$$\frac{T_1}{T_2} = e^{f\theta \frac{d}{d + t(1 + f^2)}}$$

Now, of course, so long as so little is known about the value of the coefficient of friction under various conditions, it would be out of place to make use of this more exact formula in actual calculations; but the same is nevertheless of considerable interest as it to some extent will explain the well-known, but to my knowledge never before explained, fact that, under otherwise equal conditions, a belt will slip more readily on a pulley of small diameter than on one of larger diameter, but with the same arc of contact.

Mr. Wilfred Lewis.—As the purpose of this paper is to open up discussion, some criticism of the author's conclusions is evidently invited, and appreciating, as I do, his efforts to establish new truths from old data, I would be glad to agree with him in every particular, but in this hope I have been disappointed, and it is therefore my privilege to dissent and give reasons.

It is not at all surprising that the sum of the tensions in horizontal belts should increase with the load transmitted, and this fact argues no peculiarity in leather as a transmitting medium, because it must be true of any material. This much

could have been proved without recourse to experiments, because the slack side can lose in tension only so much as is due to the length which it gains from tightening the other side. It could not lose all its tension if the belt were clamped to the pulleys and the tension on the tight side were increased to several times the original sum of the tensions.

The length between pulleys is therefore an important factor, and the tension on either side might be estimated from the catenary, which its length permits it to take. There is, therefore, no law to be anticipated as applicable to all belts without regard to position or distance between pulleys.

The surprising fact noted in the Sellers experiments was an increase in the sum of the tensions upon vertical belts. Here there was no sag between pulleys to account for it, and the explanation could only be found in a peculiarity of the belt material. The cause was immediately sought in some variable rate of extension under increasing loads, and experiments made at the time but not included in the published record fully confirmed this view.

The modulus of elasticity for leather was found to increase with increasing tension, but it could not be definitely formulated, because time also was found to have an influence which could not be neglected.

This change in elasticity naturally affects the sum of the tensions in horizontal belts, but as the length between centres increases, its influence becomes of less and less importance.

Therefore, while agreeing with Mr. Aldrich's first conclusion, it seems to me important to note the causes operating under different conditions to increase the sum of the tensions, and the impossibility of formulating the increase without regard to these conditions.

So also with the second conclusion. The third conclusion is partly true for all belts, whether vertical or horizontal; but the amount of increase in $T_1 - T_2$ and in $T_1 + T_2$ has a very decided limit for the former condition, whether the belt be long or short, while its limit for the latter condition depends chiefly upon the length of the belt.

In regard to the fourth conclusion, that the ratio of the tension cannot be calculated with any degree of accuracy by the usual formula $\left(\frac{\tau_1}{\tau_2} = e^{f\theta}\right)$, there is some ground, I think, for believing

that the assumptions upon which this formula is based are not always realized in practice. It assumes with good reason that friction is proportional to normal pressure, and increases from the slack to the tight side of the belt as it wraps around the pulley, the coefficient f remaining constant throughout, but in extreme cases, where f is very high, there is always a certain amount of gum-like stickiness on the belt surface which may vitiate the correctness of this fundamental assumption.

This gummy material in the belt, or on its surface, warms up by excessive slipping into a lubricant which finally destroys adhesion, and allows the belt to slip off. I do not think, however, that under ordinary working conditions the formula in question can be much at fault.

However this may be, whether friction is strictly proportional to the normal pressure against the pulley surface or not, the ratio $\frac{\tau_1}{\tau_2}$ must depend upon friction for existence whenever it differs from unity, and yet it is quite possible to admit in the fifth conclusion that there is a definite relation between the ratio of the belt tensions and the ratio of their sum and difference, which relation is independent of the coefficient of friction. This is the obvious relation between the sum and difference of any two quantities, but the attempt to formulate it is, in the present case, a demonstrable failure.

The proposed equation $\frac{\tau_1}{\tau^1 - \tau_2} = e^{\frac{2\tau_2}{2\tau_1 - \tau_2}}$ begins and ends in the same terms without any foundation to stand on, and to find the relation $\frac{\tau_1}{\tau_1 - \tau_2}$ we must substitute in the second term of the equation the values of the very quantities τ_1 and τ_2 that we seek to find in the first. But if τ_1 to τ_2 are known, the first term is known, and of what use, it may be asked, is the second?

In this equation a relation is set up between identical quantities, and the base l stands ready to be raised to a power which implies full knowledge of the relation sought. There is, therefore, no occasion to perform the operation, but if we do go to that trouble, the error in the supposed relation is at once established.

No matter what the coefficient of friction may be, τ_2 must be equal to or less than τ_1. If these quantities are equal, the first

term of the equation becomes $\frac{\tau_1}{0}, = \infty$, and the second, $e^{\alpha} = 7.4$, making $\alpha = 7.4$, which is manifestly absurd.

If $\tau_2 = .9\tau_1$, the first term equals 10 and the second term becomes $e\frac{1.8}{1.1} = 5.16$. The two terms of this alleged equation will approach each other in value as τ_2 grows less, but they will never accord until τ_2 becomes zero, which applies only to an impossible assumption.

The process by which this relation was deduced is clearly a case of reasoning in a circle.

Figs. 39 and 40 are legitimate expressions for interesting relations derived from experiments.

Fig. 41 graphically represents theoretical values of f for variations in the ratio of the belt tensions.

Fig. 42 assumes to be, and no doubt was, derived, as stated, by plotting a series of values from the Sellers experiments, but it is not necessarily a diagram derived from or dependent upon experiment at all. The ordinates and abscissas in this curve are mutually convertible, for when one is known the other is positively fixed, and the diagram really tells nothing.

The same may be said of Fig. 43, which simply portrays relations between different groups of identical quantities.

The relations between these different forms of the same quantities have been driven in a circle to the final result, in which the equated values may not meet, as already shown, by as much as the difference between 7.4 and infinity.

The problem to be solved in the use of belting is not so much what a given belt can pull as what it ought to be required to pull, and on this point the data given by Mr. F. W. Taylor in his "Notes on Belting" * are particularly valuable. It is really a question of dollars and cents; the larger the belt used for a given duty, the greater the first cost and the less the cost of maintenance, and we must determine as best we can how far it pays to increase the first cost for the sake of the saving in maintenance.

Mr. Taylor has shown by years of careful cost records that it does not pay to drive belting beyond a certain working limit, which is far within the bounds of common usage.

It is hard to come down to the values he recommends, because

* *Transactions* American Society Mechanical Engineers, vol. xv., No. 558.

the first cost is an obstacle in the living present, while the maintenance belongs to the dim future; but if we come anywhere near his figures, it is safe to say that the increase in the sum of the tensions may be left to take care of itself. The greatest increase is to be expected on long horizontal belts, and these are unquestionably the best suited to take it, because they can stretch more than any others without requiring attention.

The practice of William Sellers & Co., Incorporated, in the use of belting is shown by the diagrams herewith (Figs. 44 and 45). This was adopted years ago as the result of the best light obtainable, a decided difference being then made between laced and cemented belts, but with the recent improvements in lacings the difference shown may now be too great.

The working strength of the leather is probably greater than Mr. Taylor would approve, but it is on the side toward which people naturally lean, saving in first cost, and it is offered in this connection as representing in concise form the outcome of the labors of William Sellers & Co., Incorporated, on the subject of belting.

Mr. Barth.—I would like to say a few additional words in defence of analytical methods and treatment. You certainly want as complete a mathematical analysis as you can possibly get in connection with any mechanical problem, even if this leads to a formula which is too long and cumbersome for the practical engineer to make direct use of, or to one whose refinements are lost, in its application, by the indeterminateness of some assumed constant, as in this case the coefficient of friction between the belt and the pulley. The importance and possibilities of mathematical analysis are sadly unappreciated by even many of our most reputable practical engineers, who still stick to the idea of *making experiments* and *testing to destruction*. But experiments undertaken without the guidance of at least a fairly complete theory founded on analysis, more frequently lead to a misinterpretation of the observations made than to correct conclusions; while the testing to destruction of anything but a simple centrally loaded member, cannot possibly furnish you with information that will enable you to determine a proper working load. For is it not a fact that you want to limit the stresses under actual working conditions to something certain? And when you test a thing to destruction you absolutely upset the conditions and relations that exist before any material deformation takes place. Only by

FIG. 44.

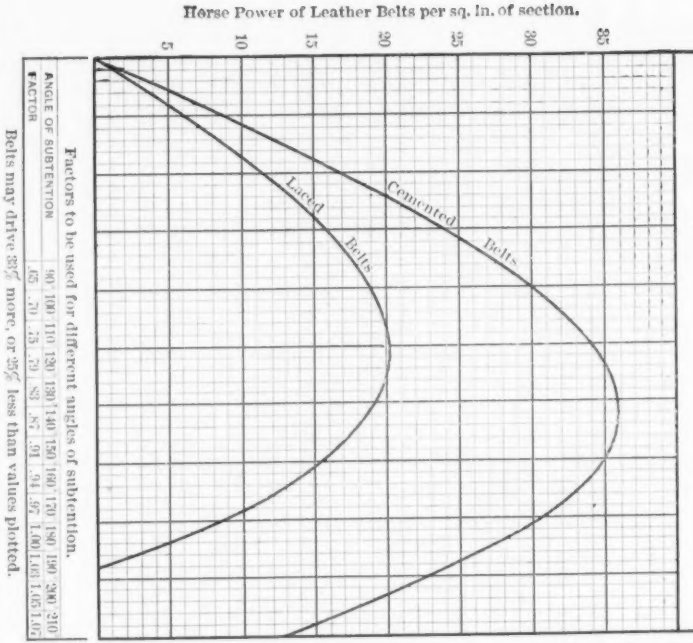
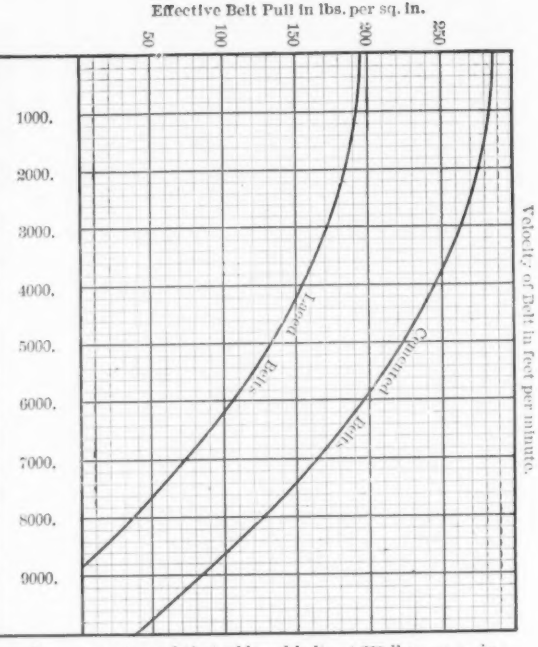


FIG. 45.



THICKNESS OF BELTS		
WIDTH	SINGLE	DOUBLE
1"	3/32	5/16
2	3/16	5/16
3	3/32	3/8
4	3/32	3/8
5	3/32	3/8
6	3/32	3/8
7		3/8
8		3/8
9		3/8
10	3/16	3/8
12		3/8
14		13/32
16		
18		
20		7/16
22		
24		
27		
30		
36		

Factors to be used for different angles of subtention.

ANGLE OF SUBTENTION	90	100	110	120	130	140	150	160	170	180	190	200	210
FACTOR	.65	.70	.75	.79	.83	.87	.91	.94	.97	1.00	1.03	1.05	1.07

Belts may drive 30% more, or 25% less than values plotted.

In these diagrams, the working strength of cemented belts has been taken at 400 lbs. per sq. in. and that of laced belts at 275 lbs. per sq. in. The angle of subtention has been assumed to be 180°, and the coefficient of friction .40 The latter may vary from .25 for new or dry belts, to 1.00 or over for belts well dressed and in the best working condition. Total pull on shaft = 1.8 effective pull of belt.

Aldrich

a judicious combination of tests and analysis can you hope to get proper proportions into your engineering structures, and you may feel sure that the possibilities of mathematical analysis in engineering are as yet far from exhausted, even though some may fancy there is nothing in it.

Mr. Oberlin Smith.—I would like to ask whether any gentlemen here have had any experience of the relative power wasted with double width and single width by the extra stiffness in belts. I want to know the power which is wasted by bending a belt which probably would bend about as the square of its thickness over the power used by a thin one.

*Prof. Wm. S. Aldrich.**—It has been repeatedly shown, and again noticed in this paper, that the most readily obtained value in any case of belt transmission is the difference of the belt tensions. In particular, in the case of dynamos and motors, the power transmitted by the belt is very easily obtained from the known efficiency of the machine, and the watts output or input, as the case may be.

If, now, to combine with this difference of belt tensions we can get another—an empirical relation between the belt tensions not involving directly the coefficient of friction, or the slip of the belt—we shall then be able to determine the separate tensions. This cannot be more than an approximation, in any case. It will be sufficiently accurate, however, to be useful in forming some idea of the sum of the tensions; that is, of the pressure on the journals. In many cases, as in dynamo and motor working, this is a desirable quantity. There are very few such instances where it can be readily measured, as by a spring balance.

The author has attempted to derive such a relation as indicated above, from a consideration of the nature of the curve shown in Fig. 42 of the paper. This curve is derived from and is dependent upon the values for the belt tensions given in 278 out of 330 of the experiments reported by Mr. Lewis. It represents the characteristic variations between the tensions indicated.

The value of the constant (k) and the resulting equation (5) have been obtained by the most thoroughly recognized process in experimental engineering; namely, by plotting the experimental results, and then proceeding to find a mathematical relation, as an equation of condition, which, with suitable constants, will apply

* Author's closure, under the Rules.

to the curve plotted. The curve, Fig. 43, was derived from Fig. 42 by the well-known method of changing the origin of coördinates; but this does not change its value in the least as representing the relations between the experimental values. It shows a family likeness to a well-known mathematical curve, the equation of which has long since been established, equation (1).

An analysis of all of the experiments on belting reported by Mr. Lewis shows that the majority of them (278 out of 330; that is, 84.3 per cent.) were run under conditions which made the ratio of the belt tensions $\frac{T_2}{T_1}$ less than 0.3. The value of the constant (k) adapted to this range was found, as stated, to be of the form given in equation (3). It is to the large number of experiments, falling within this range of values, that formula (5) under discussion is especially applicable. Table I. shows its degree of approximation for assigned values of the ratio of the tensions. It will be seen that the approximation is sufficiently close for all practical purposes for all of the experiments in which the ratio of the tensions is 0.3, or less. This includes the majority, 84.3 per cent., of all of the experiments. The two terms of equation (5) accord, therefore, sufficiently close, long before the tension T_2 reduces to zero.

It was not claimed in the paper that this formula (5) will hold true for all ranges of loads for all cases of belt transmission. It will, however, apply with a fair degree of approximation, to the experiments, eighteen in number, in which the ratio of the tensions is between 0.3 and 0.4. Beyond this latter ratio the divergence gradually becomes greater; but this includes only thirty-four out of 330 of the experiments to which the formula will not apply.

The application of the formula may be further examined by inserting, in equation (5), the values of T_1 and T_2 from any set of experiments to which the formula will avowedly apply; that is, to those cases in which the ratio of the belt tensions is about 0.3 or less. The results of such substitution are shown in Table II., for the rather extreme conditions cited in the paper, namely, from one with a coefficient of friction of 0.228 and a belt slip of 0.8 foot per minute to one with a coefficient of friction of 1.37 and a belt slip of 54.8 feet per minute.

No belt formula will apply to conditions which exist only when the belt is at rest. The values of T_1 and T_2 are then, and only

then, equal. The formula was clearly not intended for such conditions. Even the well-known belt formula, for the ratio of the tensions, is useless for these conditions. For then this ratio is of course unity, and there is no coefficient of friction. The whole problem is completely indeterminate when the belt is at rest. In fact, there is then no problem at all. You know that the two tensions are equal, and no formula is required to ascertain this.

Equation (5) was not derived to suit a single, isolated, and extreme case, such as when $T_2 = 0.9\ T_1$. There is one and only one such case in the whole list of 330 experiments reported by Mr. Lewis. The equation was derived to suit the general average conditions of 278 cases, however, which it does with a very close degree of approximation, and more closely than any other empirical formula.

TABLE I.

Showing the degree of approximation of the formula for assigned values of the ratios of the tensions.

Calculated Values for Assigned Values of $\frac{T_2}{T_1}$		Assigned Values of $\frac{T_2}{T_1}$	Number of Experiments reported by Mr. Lewis, whose actual values of the ratio $\frac{T_2}{T_1}$ fall between assigned values of this ratio.
$\frac{T_1}{T_1 - T_2}$	$\frac{2\ T_2}{2\ T_1 - T_2}$		
Infinity.	7.3891	1.0	Condition for belt at rest.
			1 experiment between 1.0 and 0.9.
10	5.1346	0.9	1 experiment at 0.9.
			4 experiments between 0.9 and 0.8.
5	3.7935	0.8	
3.3333	2.9359	0.7	
			5 experiments between 0.7 and 0.6.
2.50	2.3563	0.6	
2.00	1.9477	0.5	
			19 experiments between 0.5 and 0.4.
1.6666	1.6487	0.4	
			18 experiments between 0.4 and 0.3.
1.4286	1.4233	0.3	
			45 experiments between 0.3 and 0.2.
1.2500	1.2492	0.2	
			79 experiments between 0.2 and 0.1.
1.1111	1.1111	0.1	
			154 experiments below 0.1.

TABLE II.

Showing the application of the formula to a set of experiments with a very wide range of values of the coefficient of friction and the slip of the belt. The observed values are from Table II. of the paper by Mr. Lewis on "Experiments on the Transmission of Power by Belting."

No. of Experiment.	Tensions Measured during Test.		Derived Values of Individual Belt Tensions.		Ratios of Tensions.		Comparison of Observed with Calculated Values.				Range of Running Values of	
							Values from Observed Data.		Calculated Values of $\frac{2T_2}{2T_1 - T_2}$	Diff. between Obs. and Calc. Values of $\frac{2T_2}{2T_1 - T_2}$	Coefficient of Friction.	Velocity Belt Slip. Ft. per Minute.
	Sum of Tensions: $T_1 + T_2$	Working Tension: $T_1 - T_2$	Tight Side: T_1	Slack Side: T_2	$\frac{T_2}{T_1}$	$\frac{T_1 + T_2}{T_1 - T_2}$	$\frac{T_1}{T_1 - T_2}$	$\frac{2T_2}{2T_1 - T_2}$				
149	105	40	72.5	32.5	.449	2.625	1.813	.5777	.5950	−.0173	.228	0.8
150	110	60	85	25	.294	1.833	1.416	.3148	.3178	−.0030	.396	1.2
151	120	80	100	20	.200	1.500	1.250	.2222	.2231	−.0009	.524	1.6
153	150	120	135	15	.1111	1.250	1.125	.1176	.1178	−.0002	.723	2.8
155	182	160	171	11	.0643	1.137	1.069	.0665	.0667	−.0002	.913	4.8
156	202	180	191	11	.0576	1.122	1.061	.0593	.0592	+.0001	.950	12.0
157	216	200	208	8	.0385	1.080	1.040	.0392	.0392	.0000	1.12	23.2
158	232	220	226	6	.0266	1.055	1.027	.0269	.0266	+.0003	1.29	28.0
159	252	240	246	6	.0244	1.050	1.025	.0247	.0247	.0000	1.32	39.2
161	292	280	286	6	.0210	1.043	1.021	.0212	.0208	+.0004	1.37	54.8

DCCXCVIII.*

COOLING TOWER AND CONDENSER INSTALLATION.

BY J. H. VAIL, PHILADELPHIA, PA.

(Member of the Society.)

THE conditions existing prior to the installation of the plant referred to in the following paper were as follows:

The electric light station was equipped with 27 boilers 48 inches in diameter, 20 feet long, with 22 5-inch tubes. These boilers were set two to a furnace, grate surface 8 feet 6 inches by 5 feet, the odd boiler being set to a single furnace. The engine capacity and the load on the station already taxed all the boilers to the limit of their steaming capacity.

Plans had been prepared and estimates made for enlarging the building and increasing the boiler capacity, prior to the time the proposition was given in charge of the writer.

After a preliminary investigation of the existing conditions, the writer recommended that by the putting in of a cooling tower and a condensing system the engine capacity of the station could be increased, leaving the boiler capacity the same, thus saving the cost of adding more boilers and enlarging the building, and at the same time obtaining better economy, as well as a greater capacity for production.

To determine the steaming capacity of the boilers, two tests were made with one pair of boilers, which showed the following results under regular working conditions:

Duration of test, five hours.
Coal, Shenandoah pea.
Average temperature boiler feed, 206.5 degrees Fahr
Grate surface, 43 square feet total.
Average steam pressure, 97.4 pounds.
Average temperature boiler room, 69 degrees.
Average temperature stack gases, 440 degrees.

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* Presented at the New York meeting (December, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

Coal fired, 6,280 pounds.

Weight of ashes, 892 pounds.

Weight of combustible, 5,388 pounds.

Water evaporated, 704.6 cubic feet, 42,818 pounds.

Water evaporated per pound of coal from and at 212 degrees, 7.13 pounds.

Water evaporated per pound combustible from and at 212 degrees, 8.24 pounds.

Coal burned per square foot of grate per hour, 29.2 pounds.

This test showed that each 48-inch by 20-foot boiler would evaporate into steam 4,281 pounds of water per hour, giving a capacity of 115,587 pounds steam per hour with every boiler in operation. Coincident with the boiler test, one 18½ by 30

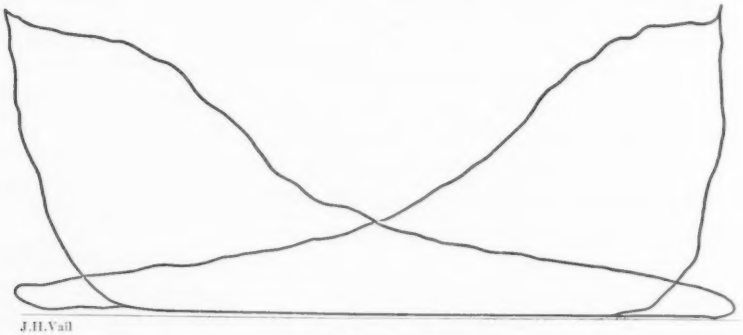


FIG. 46.

Buckeye engine was using the steam from the boilers under test. Indicator cards were taken at fifteen-minute intervals for five hours. The result from the engine test and average of all cards showed a steam consumption of 46 8 pounds steam per indicated horse-power per hour. A pair of specimen cards are shown in Fig. 46, and will serve for comparison with cards taken from the same engine some months later, after alterations.

After the data derived from these tests had been analyzed, the writer recommended that the Buckeye engine should be converted from the 18½ by 30 high-pressure engine into a 14¼ and 25 by 30 tandem compound condensing engine; also that an additional 750 horse-power tandem compound condensing engine should be erected in the station, together with a cooling tower and the necessary condenser equipment, and that the only change in boilers should be to raise the working pressure. No increase of boiler capacity has been made.

After investigation, the Barnard Patent cooling tower was selected as desirable to best meet the conditions existing at this plant, which were minimum floor space, and minimum weight, and a considerable elevation above floor level of engine room.

The elevation and general view of the tower are shown in Fig. 47, while Fig. 48 gives the interior arrangement of mats, etc.

Details of Cooling Tower.

The cooling tower is of the twin type, having two chambers, with a pair of fans supplying a strong draft of air to each chamber. The interior dimensions are 12 feet 3 inches by 18 feet by 29 feet 6 inches high.

The tower is mounted on a substructure of steel columns and I beams, supported on substantial foundations.

There are outside galleries and ladders affording convenient access to necessary points.

The shell of the tower is of steel plate, properly reënforced with angle and channel irons.

The hot water from the condenser discharge is delivered through a 10-inch wrought-iron pipe, extending the whole length of each chamber, and is slotted on top and perforated at the bottom, giving equal distribution to a series of distributing pipes extending across the tower, each pipe being slotted and perforated, thus insuring a very uniform distribution of water.

Means are provided for cleaning these pipes, which is found necessary in cold weather, when the cylinder oil from the exhaust steam is liable to clog the pipes and interfere with the uniform and free distribution of the water.

The hot water falls from the distributing pipes over galvanized wire mats, arranged as shown in Fig. 48.

Each mat is suspended by galvanized iron hooks, and is easily removed for cleaning or repairs.

In actual service it is found that the water is uniformly distributed.

The circulation of air is furnished by two pairs of steam-driven fans, each pair of fans being mounted right and left on the same shaft, and the four fans being capable of delivering the necessary quantity of air when driven at a speed not exceeding 150 revolutions per minute.

The air entering the tower chambers at the lower section is

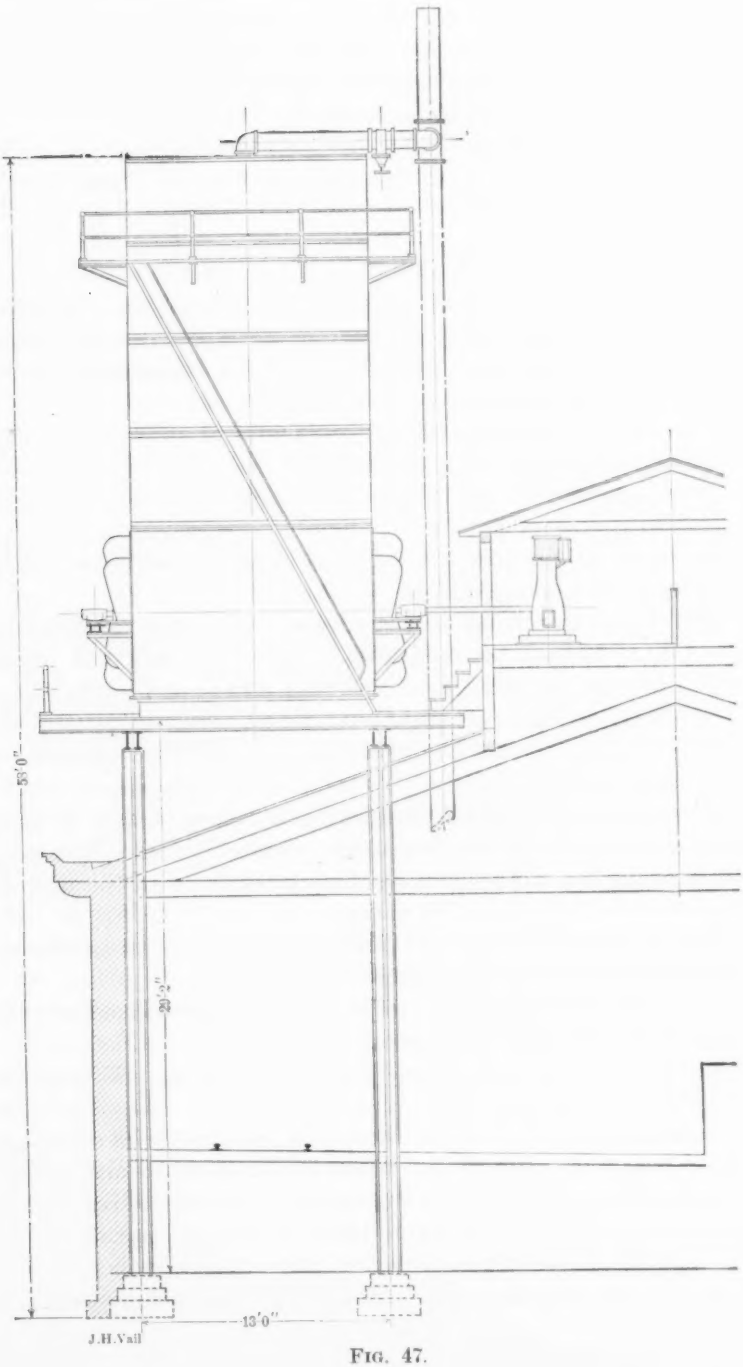


FIG. 47.

deflected vertically from each fan, thus avoiding cross currents, and affording a uniform blast upward and between the mats.

The rated capacity of each section of this cooling tower is to cool the circulating water needed to condense 12,500 pounds of exhaust steam, from an initial temperature of 132 degrees Fahr. to 80 degrees Fahr., when the atmospheric temperature does not exceed 75 degrees Fahr., nor the humidity 85 per cent.

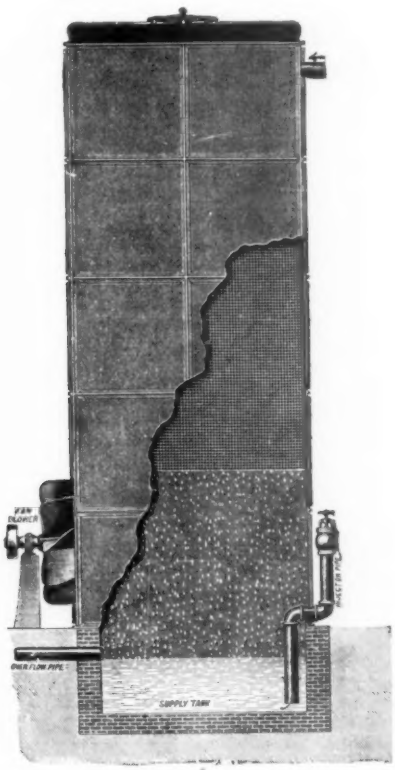


FIG. 48.

On account of suitable floor space for the cooling tower not being available near the ground level, it became necessary to mount it on a superstructure above the boiler room, the total elevation from the condenser to the discharge opening at top of tower being 58 feet.

This elevation places more duty on the air pump than is desirable, but was unavoidable for the reasons stated.

The circulating water is handled by a Blake vertical twin air pump and jet condenser (Fig. 50). Indicator cards showing the performance in the steam cylinders of the air pump are shown in Fig. 49.

In an equipment of this kind it is very important to have facilities for driving the fans at variable speeds; this requisite flexibility has been obtained by using a small vertical engine, direct connected to the shaft of each pair of fans.

The first intention was to drive the fans belted from electric motors. An analysis of all the conditions proved this method

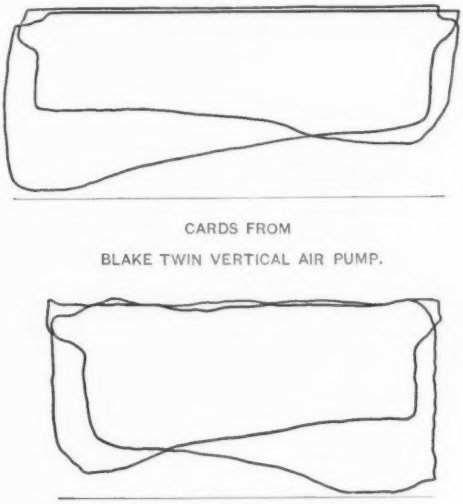


FIG. 49.

undesirable for the reason that there would be too many translating devices interposed between the live steam pipe and the shaft of the fan. These translating devices would have been as follows:

The engine, the generator, the transformer to reduce the primary current to secondary current for use on the motor, a motor with intermediate shaft, belting and pulleys to reduce the speed to the fans, and the losses in intermediate devices would more than overcome the original low cost of power.

In this connection it must be remembered that two-phase A. C. current being the only kind available, there is a notable disadvantage in the induction motor; that is, the speed is not

subject to wide variation, and any special means of external regulation to vary the speed would tend to reduce the efficiency. It was, therefore, determined to direct connect with the shaft of each fan a vertical engine without a governor. This plan has proven to be exactly what is required in practice, and, moreover, its first cost was only one-fourth of the cost of a motor equip-

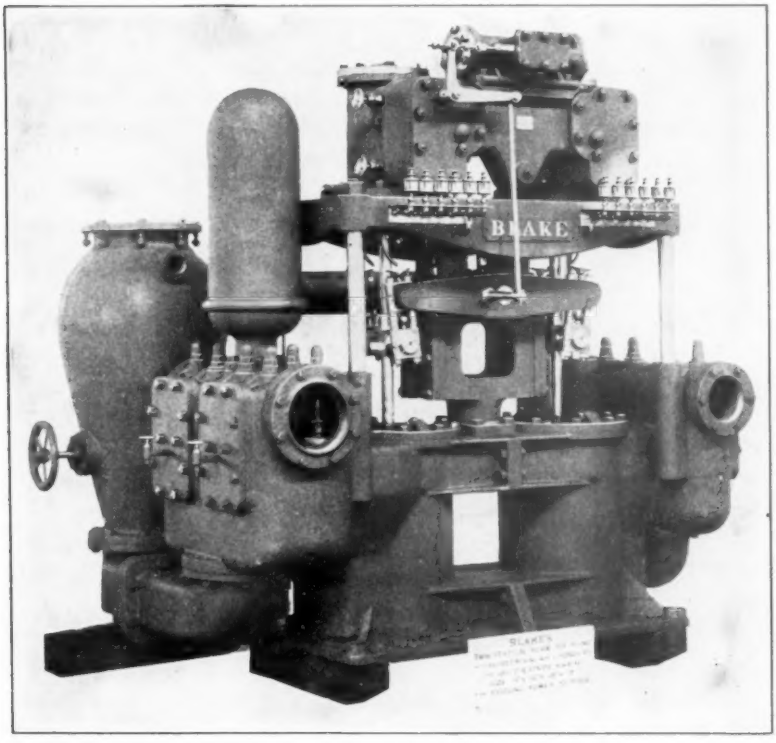


FIG. 50.

ment, with lower cost of operating expenses also largely in favor of the engine driving feature.

Under varying conditions of temperature, the speed of the fans must be increased or decreased, according to high or low atmospheric temperatures. In winter there are many hours during each day when the low temperature of the air circulating through the cooling tower will cool the water for a partial engine load without running the fans at all. During high temperature in summer the fans must be run at maximum speed.

The following table, extracted from the log records for many months, shows details as to temperatures, speed of fans, reduction of temperature of condenser discharge, etc.

Table of extracts from log-book showing temperatures, vacuum, speed of fans, heat extracted through cooling tower, etc.:

	1898.					
	Jan. 31.	Feb.	June 20.	July.	Aug. 26.	Nov. 4.
Time	9 P.M.	8 P.M.	8 P.M.	8 P.M.	8 P.M.	5.35
Temperature atmosphere	30°	36°	78°	96°	85°	59°
" condenser discharge water to cooling tower	110°	110°	120°	130°	118°	129°
Temperature injection water returned from tower	65°	84°	84°	93°	88°	92°
Degrees of heat extracted by tower	45°	26°	36°	37°	30°	37°
Speed of fans at tower, R. P. M.	36	0	145	162	150	148
Vacuum at condenser, inches	25½	26	25	24½	25½	25
Strokes of air pump	30	30	37	44	43	28
Boiler pressure, lbs.	110	110	120	120	120	112
Temperature boiler feed	212°	212°	210°	211°	213°	213°

As previously noted, the 18½ by 30 Buckeye engine was changed to a 14¼ and 25 by 30 tandem compound condensing engine by bolting new tandem cylinders on the existing frame and making necessary alterations in valve rods, etc.

Specimen cards taken from this Buckeye engine after being compounded are shown in Fig. 51. In referring to these cards please note the following data:

Revolutions, 137; steam pressure, 113 pounds; mean effective pressure, 50.16 pounds; vacuum per gauge, 26 inches; horse-power developed in high-pressure cylinder, 163.42; horse-power developed in low-pressure cylinder, 168.48—total, 331.9 horse-power, and of this 90.52 horse-power is below atmospheric line. It will be noted that the work is divided almost equally between the high-pressure and low-pressure cylinders; all cards show similar results.

This change in an existing engine clearly illustrates the advantage derived from the condensing system. This engine was fitted with a receiver and reheater between high and low-pressure cylinders; several tests were made, and representative cards were analyzed to determine whether the reheater was of value, and as a result of these tests we found that the reheater condensation amounted to 63 pounds of steam for each

horse-power gained in the low-pressure cylinder. We therefore abandoned it as an expensive luxury. We are not now discussing the value of reheaters, and for this reason I simply give the facts without entering into details.

In addition to the tandem Buckeye engine, a tandem compound

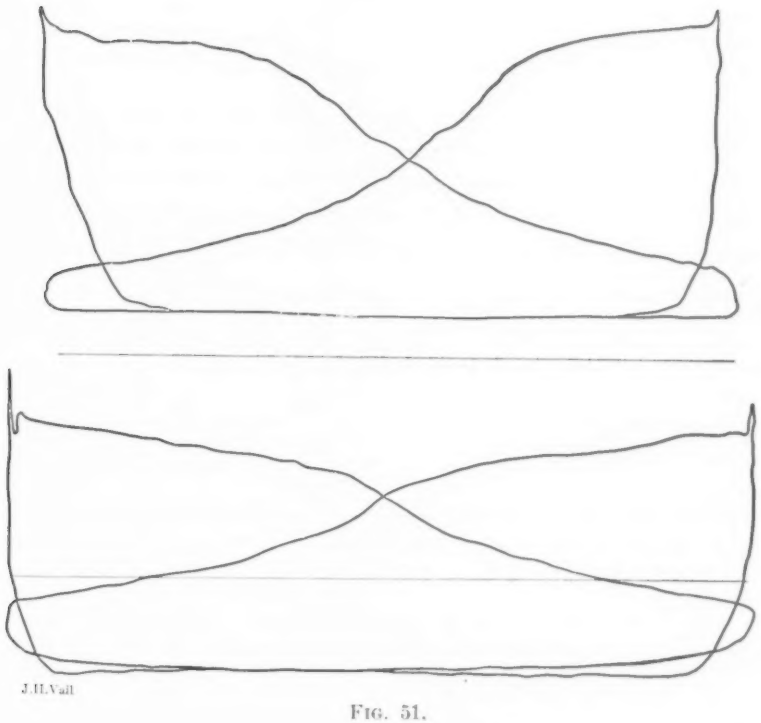


FIG. 51.

condensing engine, 20 and 36 by 42 inches, 120 revolutions per minute, Corliss type, built by Pennsylvania Iron Works, was installed to drive a direct connected Stanley 500 kilowatts two-phase A. C. generator. This engine works fifteen to seventeen hours per day. Specimen indicator diagrams are shown in Figs. 52 and 53, which clearly indicate the advantage of the condenser service.

The usual work required from the cooling tower and condenser varies from 7 to 17 hours per day. A notable record was made on August 2, 1898, when the run was from 7 A.M. till 12 midnight; and from the daily records the following data are extracted:

	Maximum.	Minimum.
Temperature atmosphere	103°	83°
Temperature condenser discharge water to tower	128°	106°
Temperature injection water from tower	98°	91°
Degrees of heat extracted, by tower	32°	21°
Speed of fans, revolutions per minute	160	140
Vacuum at condenser, inches	26	26
Strokes of air pump	50	38
Boiler pressure, lbs	121	100
Temperature boiler feed	212°	200°
Engine horse-power developed	900 H.P.	400 H.P.

A continuous heavy load was carried during the entire 17 hours' run. This was not a test record, but simply daily service.

Fig. 53 shows indicator diagrams taken November 5, 1898, from 20 and 36 by 42 tandem compound condensing Corliss engine. The conditions were as follows:

Engine revolutions per minute	120	
Steam pressure, lbs	112	
Vacuum at condenser, inches	25	
The area of the cards shows the work done in high-pressure cylinder to be		311.8 H.P.
And in low-pressure cylinder		231.5 H.P.
Total I.H.P.		643.3 H.P.

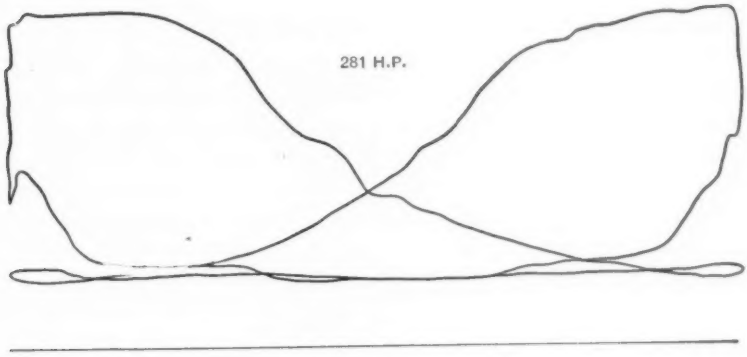
Work done in low-pressure cylinder below atmospheric line, 185.1 horse-power. Simultaneously with the engine, the air pump and fan engines were indicated.

The work done by the air pump	13.75 H.P.
The work done by the fan engines	13.5 H.P.
Total external work	27.25 H.P.

The amount of work, if deducted from the work done below atmospheric line in low-pressure cylinder (*i.e.*, 185.1 horse-power), leaves a net gain of 157.85 horse-power by the use of the condenser and cooling tower.

It will be noticed from the previous data that the feed water shows a temperature above 200 degrees Fahr.

There are two feed-water heaters in connection with the condensing plant. First, an intermediate tubular heater in the line of exhaust between low-pressure cylinders and condenser. Second, an auxiliary feed-water heater was also attached, receiving the exhaust from the condenser and boiler-feed pumps and any other auxiliaries.



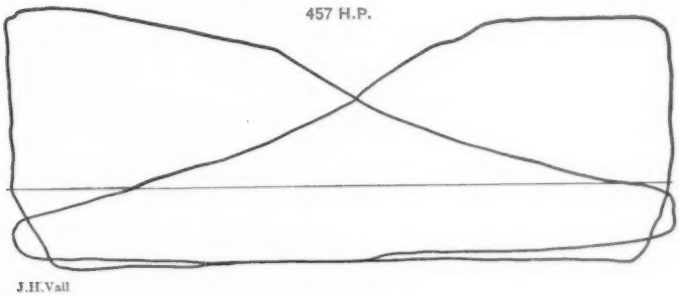


FIG. 52

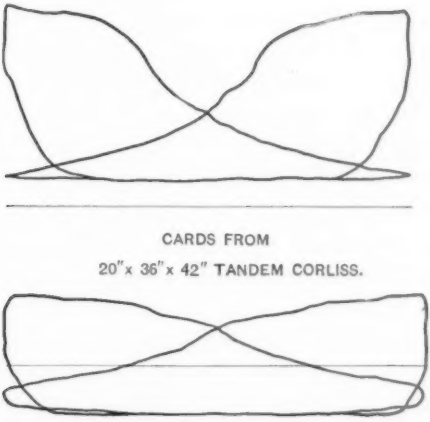


FIG. 53.

The feed water is first heated in a tank that receives the exhaust from the general line of high-pressure engines. The feed water then passes through the intermediate heater, and thence through the auxiliary heater, and reaches the boiler at a temperature upwards of 200 degrees Fahr.

FIG. 54.

The pressure of other important work has prevented the writer from making accurate tests to determine the exact economy derived from the condensing plant, and while awaiting the opportunity to make such tests, we have the satisfaction of having increased the station capacity about 1,000 horse-power with the aid of a condensing system, using the same water in continuous circulation, and also that the boiler plant, previously stated to be fully loaded, now supplies steam for this additional work, with boilers to spare.

DISCUSSION.

Mr. F. Meriam Wheeler.—The subject of cooling water by mechanical means for use in condensers is attracting considerable attention in this country and abroad. I regret very much that Mr. Vail has been prevented from being present, as no doubt some of the members would like to ask him questions. He writes me that pressure of time prevented him from making his paper as complete as he would like to have done in regard to the details of the cooling tower and pumping plant. He also says in his letter: "It may be of interest to the meeting in connection with the subject of my paper to learn that a few days ago the 750 horse-power condensing engine, referred to therein, was operated at half load for four or five hours, at which time the fans in the cooling tower were *not* in operation—the atmospheric temperature being from 38 to 40 degrees. We were able to carry a vacuum on the condenser of 26 inches. The heated injection water was pumped from the condenser to the tower at a temperature of 116 degrees, and after passing through the cooling tower was reduced

to a temperature of 90 degrees. This we find is about the limit of load for the cooling tower when the fans are not in use."

Now, in regard to the Blake air pump used in this installation, and shown by Fig. 50, it is of peculiar construction—probably different from anything of the kind heretofore used. The design is quite novel, and is the first of its kind installed in this country. It is not only an air pump in connection with a jet condenser, but is also a water-elevating pump—that is to say, it elevates the injection water (after it is used in the condenser) to a height of about 58 feet; *i.e.*, the distance from where the condensing apparatus is located to the top of the cooling tower. To use an ordinary air pump to lift to such a height would, of course, vitiate the vacuum very much, owing to the compression of the vapors in the air cylinders; furthermore, it would be a very expensive method of lifting the water. I never recommend an air pump to have more than 6 feet head over its discharge valves. In fact, I would prefer not to have more than 6 inches—just enough for sealing the valves.

The problem in this case was to operate an air pump and jet condenser in connection with a Barnard cooling tower, where the water had to be pumped to a height of 58 feet, without vitiating the vacuum, and yet have a fairly economical method of lifting the water to the elevation named. This problem was successfully solved in the following manner: I divided the pumping engine into two elements, *i. e.*, a so-called "wet vacuum" air cylinder and a "dry vacuum" air cylinder, the former taking solid water from the bottom of the condenser, while the latter pumped off the non-condensable vapors and air from the upper part of the condenser. The vapors and air as they pass to the dry vacuum air cylinder are cooled by a small jet of cold water (not shown in the engraving) sprayed into the vapor pipe. I originally provided a $\frac{3}{4}$-inch water connection for this jet, but it was found unnecessarily large, and was replaced by a $\frac{3}{8}$-inch connection, the valve of which was only required to be open a small amount. I mention this to show how little water is necessary to properly cool the vapors, the temperature of which was about 78 degrees at the discharge.

By this unique arrangement of air pump we can hold as high a vacuum as 28 inches, and the report of last week's running shows an average of 27-inch vacuum. The arrangement of the steam end, it will be noticed, consists of two steam cylinders, the same as used in the regular Blake vertical twin air pump arrangement,

together with two air cylinders, each of which is double-acting, to which the jet condenser and vapor pipe are connected as above described. It is surprising to note how much air comes off from an air pump—much more than the average engineer has any idea of. One of the troubles with engine builders and others is that they do not look more carefully for air leaks. Often an engineer will insist that everything is tight, while it is afterwards found to be the very opposite. An air pump of this kind is a telltale, as one air cylinder is employed solely to pump off the air and non-condensable vapors.

Mr. Vail explains in his paper why the tower was placed at such a high elevation. Now, if the cooling tower was placed on the ground, the air pump would have had to lift only one-half the height, hence the indicated horse-power would approximate but half of what is shown.

Mr. Rockwood.—Do I understand you, Mr. Wheeler, to say that these air pumps were vertical double-acting air pumps?

Mr. Wheeler.—Yes, and in that respect they are somewhat unique. Heretofore it has been generally considered impossible to make an efficient vertical air pump of the double-acting type, and also one that would do as much work on the up stroke as on the down stroke. In this design I have solved that problem quite effectively.

Mr. Rockwood.—And that $\frac{3}{8}$-inch stream of water, was it all you let in the vapor pipe to make the pump work satisfactorily?

Mr. Wheeler.—Yes. I am sorry to say that I have no knowledge as to the actual amount of water. I simply repeat it was a $\frac{3}{8}$-inch valve partially opened, say about three-fourths of a turn. The amount of water used was certainly very small.

The "wet vacuum" cylinder performs no other service than taking water from the bottom of the jet condenser and forcing it to the top of the tower, and is supposed to travel full of water. Of course, if they allow the pump to run at a greater speed than necessary, the piston of the "wet vacuum" cylinder will leave the water. In other words, the cylinder will not fill completely, and not accomplish what was intended. It will be noticed that we have provided our safety float arrangement, which is enclosed in the round part projecting from the side of the condenser. This is for breaking the vacuum, in case of the condenser flooding, a feature which we apply to all our jet condenser air pumps, whether horizontal or vertical. In this particular case it inci-

dentally becomes a very good regulator for the speed of the pump, as the instant the water rises in the condenser above a certain height, it lifts the safety float, which, in turn, opens the little valve on the top of the casting. Of course, the inflowing air makes the vacuum drop off a little, and the load of the pump becomes less. The decreased load causes the pump to start off at a higher rate of speed, which is maintained until the pump increases the vacuum, and, consequently, the load. In fact, this device gives a perfect regulation for a vacuum pump of the direct-acting type. The steam, of course, must be kept at a steady pressure. The other pump cylinder, which we call the "dry vacuum" cylinder, takes the air and vapors from the top of the condenser, the same being cooled on the way by a small jet of water. This cylinder is also double-acting, and discharges into a tank, the air passing off, and the water utilized for feed or other purposes.

Mr. Jesse M. Smith.—Where is that small jet applied?

Mr. Wheeler.—Anywhere on the vapor pipe, usually nearer the air cylinder than the condenser.

Mr. F. A. Halsey.—Why not take that jet from the injection water and be done with it?

Mr. Wheeler.—I do not quite understand you. The injection water is sprayed into the condenser through the inside adjustable cone, then falls to the bottom and is pumped off by the "wet vacuum" (water) cylinder, and forced direct to the cooling tower. The non-condensable vapors and the air are taken off from the top of condenser by the "dry vacuum" (air) cylinder As I before remarked, the air in the vapor pipe is cooled by the small jet of cold water, hence there is no necessity of any connection with the regular injection connection. It is preferable to get the coolest water possible. In fact, the cooling jet in this installation is taken from the city main, and consequently is quite cool, whereas the injection water coming from the cooling tower, as shown by Mr. Vail's record, is from 80 to 90 degrees temperature. I hope I have made this matter perfectly plain.

Mr. William Kent.—Wouldn't it be just as well to use the 90-degree water in this secondary (vapor) pipe, for the reason that the vacuum is limited by the temperature of the water in the water cylinder—that you cannot get any more vacuum?

Mr. Wheeler.—It is very natural they should desire to use as little water as possible in the vapor pipe, hence taking it from the coolest source available. About twelve years ago I made

some very interesting tests in connection with my surface condenser, and I found that if I separated the vapors from the condensed steam and cooled them down to a low point, I secured a better vacuum than is usually obtained under ordinary practice.

Mr. Kent.—It seems to me, in the jet condenser chamber, if you had the water coming in at 90 degrees, and going out at 130 or 140 degrees, that you cannot get any higher vacuum than that due to the temperature of the outflowing water, no matter how much you cool the vapors afterwards.

Mr. Wheeler.—You are quite right, but in practice air pumps and condensers, as usually constructed, rarely give a vacuum due to the temperature of the discharge water. Referring to last week's log, just received from the electric light plant Mr. Vail refers to, I notice that at 7 P.M. on November 21st the vacuum was $27\frac{1}{4}$ inches, while the temperature of the water discharged by the "wet vacuum" cylinder of the pump was 110 degrees.

I also understand the temperature of the air discharged by the "dry vacuum" cylinder was about 75 degrees. The temperature of the injection water was 80 degrees, and the temperature of the outside atmosphere 49 degrees.

Now please note that the temperature of the discharge water was moderate enough to hold the high vacuum recorded, in fact was close to ideal conditions.

Mr. Kent.—A temperature of 110 degrees would hold that anyway. If you cooled that vapor down to 40 or 50 degrees it would not have helped things.

Mr. Wheeler.—No. It would not, but please keep in mind the fact that in practice rarely, if ever, are perfect results secured. As you know, pumps and condensers, as ordinarily used, average about 25 inches of vacuum. Theoretically this vacuum means a temperature of about 134 degrees for the discharge water. As a rule the average temperature in a hot well is about 110 degrees—quite a difference. Now, where we can separate the air from the water we make the latter more efficient. In the so-called dry vacuum leg pipe condensers, commonly used in connection with vacuum pans (whether on the parallel or countercurrent system), the air is cooled and pumped off by a separate pump, and by so doing they almost secure ideal results. Of course, the higher the temperature of the discharge water for a given vacuum, the smaller the amount of water used. As you can readily appreciate, with the cooling tower system it is very important to handle as little

water as possible. The design of the particular air pump used by Mr. Vail was to meet the demand for a compact and economical pumping plant that would lift the condensing water to a high elevation and yet maintain a first-class vacuum.

Mr. Bole.—Why not make a connection from the float in the condenser to the steam throttle valve of the air pump, so as to control the speed and run faster when necessary, without breaking the vacuum.

Mr. Wheeler.—Yes. That arrangement I think would work very well, but the arrangement we have is simpler. The very minute the little valve is opened by the float, the vacuum is reduced, and the pump speeds up to carry off the water. The pump is therefore made self-regulating.

Mr. Rockwood.—What happens to the lifting pump when it pumps the condenser dry?

Mr. Wheeler.—When it pumps the condenser dry, it would lose its speed more or less.

Mr. Rockwood.—I did not mean actually dry. I mean so that it is not taking a full cylinder of water every stroke.

Mr. Wheeler.—It would then act like any direct-acting air pump.

Mr. Rockwood.—Won't it slam?

Mr. Wheeler.—Yes, and it would act in that manner. That is why we have the regulating device described, and it keeps the cylinder full of water. Otherwise it would slam the same as any pump with its suction pipe throttled so that it did not get a full supply of water.

Mr. Jesse H. Smith.—Why did they use a jet condenser instead of a surface condenser in this installation?

Mr. Wheeler.—It is quite natural you should ask that question, for had the surface condenser system been used, they would have had the advantage of the down column of water to balance the up column, and therefore they would only have a load due to the friction in the water pipes, and the condenser tubes, plus the head due to the height of the cooling tower itself, which latter is usually about 35 feet. In other words, instead of having it pump against a head of 58 feet plus friction in the water pipes, their load would have only been equal to a head of 35 feet plus the friction in the water pipes and the tubes of surface condenser—quite a difference. I imagine the reason for not using the surface system in this case was the fact that they could not go to

the necessary expense on account of a limited appropriation. In the arrangement of the air pump and condenser described I did my best to solve the problem of lifting the water with one and the same pumping machine, with the least cost of power and without vitiating the vacuum.

Mr. Smith.—Then in your practice you prefer the surface condenser for cooling towers?

Mr. Wheeler.—Every time—it is an ideal plant for this kind of work. There are two installations of this type at the Edison Company's Twelfth Street station in this city, where the cooling towers are placed on the top of the building, and although the elevation is very high (the pumps and the surface condensers being in the basement), the power required to elevate the water is very little. Large pipes were used for both the up and down columns, reducing friction to a minimum.

Mr. Kent.—What would Mr. Wheeler think about the oil from the engine in connection with surface condensers?

Mr. Wheeler.—That is where the surface has the advantage over the jet in this cooling tower system, as you can do what you like with the water of condensation. You can go to the trouble of filtering this water for feeding to the boilers, or you can utilize it by allowing it to go to the cooling tower for condensing purposes, or, if desired, this condensed steam can be allowed to go to waste.

Mr. Kent.—It costs more for water, though, if you let it go to waste.

Mr. Wheeler.—Of course, that stands to reason. In this cooling tower system, as you perhaps know, the evaporation going on in the cooling tower is less than the amount of water fed to the boilers, so there is always a surplus to overflow.

Mr. Kent.—Some years ago we discussed the question of feed-water filters, and came to the conclusion that there were no good ones. Can you say whether there are now any satisfactory filters?

Mr. Wheeler.—I can answer that by saying: If parties will go to the expense of putting in a proper filtering plant there is no trouble at all about satisfactorily filtering feed water from surface condensers. There is a large electric light and power plant in Chicago where they have been using feed water from surface condensers for over six years, and have had no trouble whatever with their boilers; they were willing, however, to invest enough to provide filters amply large and properly constructed.

Mr. Smith.—What kind of filters did they use?

Mr. Wheeler.—They are of the quartz or gravel system. True they are quite elaborate and require considerable room, but they get the desired result and save large water bills. The trouble with most installations of filters in this country, at least in stationary engine practice, is that they have not been willing to provide filters of suitable size and construction, and the attendants have not given the proper care and attention.

Mr. Kent.—Have you had any experience with the Turkish towel system of filters?

Mr. Wheeler.—Yes.

Mr. Kent.—Is it good?

Mr. Wheeler.—This system is more adapted for marine service, where room is limited, and it needs considerable attention. Such filters require to be cleansed every day. There are several filters in the market of this kind, and they are nothing more or less than strainers of cloth, or other fibrous material. They are usually arranged for a reverse current of steam for cleaning them. The White Star boats, in fact most of the transatlantic steamers, are provided with such filters. Sometimes they first take the feed water through filter tanks having coke, sponges, salt, hay, "excelsior," or something of that kind, after which the feed water is passed through one of these towel filters. The Edmondston filter is probably the most successful abroad. In this country the use of filters for stationary engine practice has been so far rather disappointing. The average stationary engineer now being familiar with the use of feed-water filters, has not done them justice, and then again such filters that were installed of the Edmondston type were too small for stationary use. That is to say, for the same horse-power the filter for stationary use ought to be larger, and then it requires less attention. In order to use condensed steam from surface condensers, you must first have a good-sized filter, properly constructed and fitted. Even then do not try to save all your feed water. Be satisfied with saving, say, three-fourths of it; then clean your boilers at least once a week, filling up with fresh water each time, and when starting up a new plant do not try to use your feed water from condenser—just wait a few weeks. When an engine is first started up, we all know that oil is used very liberally, especially if it carries heavy loads in the beginning. This excess of oil chokes the filters; trouble with the boilers immediately ensues, and the system is promptly

(but unjustly) condemned. Now, if there was a little intelligent care at the start not to use the water from the filters until the cylinder and valve surfaces of the engine were in good condition, then all would go well. After that, it should be looked to that a minimum amount of oil is used. Engineers, as a rule, use more oil than they need to, and as to quality—the oil is frequently not fit to be used in an engine. If it were possible to use *no oil at all*, then we would have an ideal arrangement. By the way, I understand that there are many cases now where, in marine practice, no cylinder oil whatever is used—of course in such cases the engines are vertical. The auxiliary engines on many of the war vessels are also run without oil, and we are now building vertical steam pumps for torpedo boat practice, to run without oil—the same as with the main engines. Where room is so very crowded, as in the case of a torpedo boat, it is a great thing to be able to do away with feed-water filters.

DCCXCIX.*

THE THEORY OF THE MOMENT OF INERTIA.

BY C. V. KERR, CHICAGO, ILL.

(Member of the Society.)

FOUR years ago the writer presented to the Society a paper with this title.† Further consideration of the idea involved has furnished material for a more complete and useful expression of it. To offer that is the purpose of the present paper.

To avoid reference that may be inconvenient, the fundamental notions underlying this particular theory will be briefly restated here. Considering force as an action between two bodies tending to cause change in their relative rest or motion, assume the unit to be that force which, acting for one second on the unit quantity of matter, the standard pound, imparts to it a velocity of one foot per second. Since the attraction of gravity would in the same time impart to the standard pound a velocity of g feet per second, it will equal g units of force. The unit force is, therefore, equivalent to a pull or push of $\frac{1}{g} = \frac{16}{32.2} =$ ½ ounce, nearly. The unit thus defined has been called the "poundal," and corresponds to the "dyne" of the metric system, which, acting for one second on a gram of matter, imparts to it a velocity of one centimetre per second.

To impart unit velocity to w pounds will require w *poundals* $= \frac{w}{g} = m$ *pounds*. *Mass*, therefore, is the measure of the pull or push in pounds required to impart unit velocity in one second to w pounds of matter. To impart to this body in one second a velocity of v feet per second would require a pull or push of $\frac{w}{g} v = mv$ pounds. *Momentum*, then, is a measure of the force required to impart to w pounds a velocity of v feet

* Presented at the New York meeting (November, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

† "On the Theory of the Moment of Inertia," vol. xvi., p. 477, No. 629.

per second in unit time. Since a moving body coming to rest under the action of a constant force passes over a space equal to one-half its initial velocity, the *work* done on it by the retarding force will be *force* × *space* = $mv \times \frac{1}{2} v = \frac{1}{2} mv^2$, the expression for the *kinetic energy* of the moving body measured in foot-pounds.

Inertia is a property of matter enabling it to resist a change of condition as to rest or motion. In case a moving body imparts motion to a body at rest, inertia supplies both the acting and the resisting forces—the *action* and the *reaction*, always equal and opposite. And like *energy*, or capacity for work, which is indefinite until measured in foot-pounds by force through space, inertia is not a fixed quantity, but varies with the amount of matter and the gain or loss of velocity per second. *Momentum is the measure of the inertia developed.*

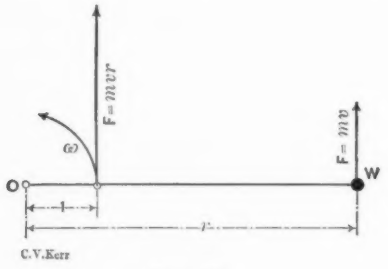


FIG. 55.

To avoid an oversight made by some readers of the former paper, v will represent linear velocity in feet per second; a, the linear acceleration or change in linear velocity per second; ω, the angular velocity of a revolving body measured in feet per second on an arc of one foot radius; α, the angular acceleration or change in angular velocity per second. Now, a may be equal to, greater, or less than v; and similar relations may exist between α and ω. Finally, $a = \alpha\, r$ and $v = \omega\, r$.

The moment of a force, F, in Fig. 55, at the end of a lever arm, r, is measured by the product $F \,.\, r$, expressed numerically as inch-pounds or foot-pounds. Consider a quantity of matter, w, concentrated at the distance r from an axis at o, whose velocity of v feet per second is to be destroyed in one second. Then its momentum, mv, will be the reaction to the force F, and its moment about o will be $mv \,.\, r$, which is equivalent to placing a single force, $F = mv \,.\, r$, at unit distance from o. Hence we may

say that the *moment of inertia* of the body w is equivalent to a force, mv, acting at a given distance, r, from a centre, o; and that it is expressed in inch-pounds or foot-pounds as convenient, which may consequently be simply added or subtracted. Here the change of velocity, or acceleration, is $v = \omega r$ feet per second, and may be replaced by $a = \alpha r$, which represents an acceleration of any magnitude, and will hereafter be used.

If a body of any form be given a rectilinear acceleration, each one of an indefinite number of parallel forces, df in Fig. 56, acting on a particle of matter, dm, will have a moment about an axis, o, of $df \cdot x = dm\, a\, x$, and the distance from the axis to the resultant of accelerating forces will be

$$s = \frac{\int dm\, a\, x}{F = \int dm\, a} \quad \ldots\ldots\ldots \quad (1)$$

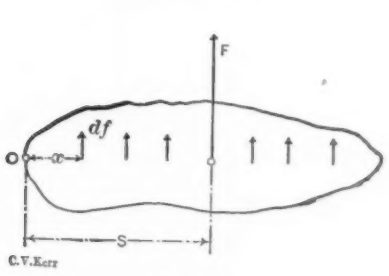


FIG. 56.

This point may with propriety be called the "centre of mass" when considered in connection with mass and acceleration. It will in general coincide with the centre of gravity, with the difference that for a given position of the body the resultant, W, of the gravity forces must act in one direction, while the resultant, F, of accelerating forces may have any direction. For convenience, however, and to conform to custom, an axis through this centre of mass will be called a "gravity axis."

Before considering the moment of inertia in its more general form, it will be of interest briefly to discuss the rotation of a body about a free axis. If we assume a body of any form, as in Fig. 57, to be rotating about a free axis at o, then any small portion will be acted on by two forces: first, the accelerating force, F, which, if directly applied as represented, will be balanced by the inertia force, F_1, and hence will not produce a reaction at the axis o; and, second, the centrifugal force, C, acting radially and balanced

by the centripetal force, C_1, or resistance of the axis. Its value is measured by $m\,\omega^2\,r$, the usual expression for centrifugal force. Now, a free axis is not supposed to be held or supported in any way, so that, if the axis o is to remain in one position while the body rotates about it, the algebraic sum of the forces $m\,\omega^2\,r$ must be zero; or, in other words, the centrifugal force of each particle must be balanced by an equal and opposite force on the other side of the axis. The axis should, therefore, pass through the centre of gravity of the body; and for the greatest stability of rotation the direction of the axis should be such that the arithmetical sum of centrifugal forces is a maximum. The reason is, that each particle under the action of centrifugal force seeks the

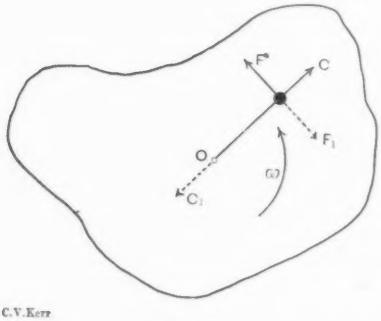


FIG. 57.

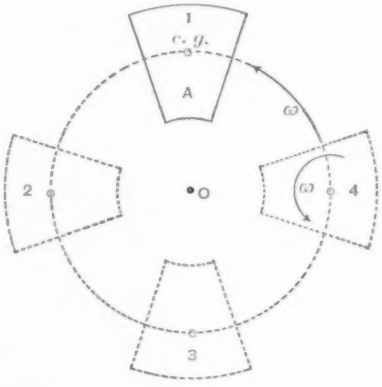


FIG. 58.

farthest possible position from the axis. This corresponds to the condition that the most stable position for a body supported at rest is that for which the centre of gravity is lowest.

If a body, A, be revolved about an axis, o, as in Fig. 58, through the positions 1, 2, 3, 4, 1, it will take in succession during a revolution all possible positions relative to the axis in the initial position; in fact, a revolution in the same time and direction, and with the same changes in angular velocity about a gravity axis parallel to the axis at o, will be made. If the body is not constrained to rotate about the gravity axis, it will remain parallel to the original position while revolving about o.

To ascertain the forces required to accelerate a body of any form about an axis, o, as in Fig. 59, and the moments of these forces, consider first the couple, df_g, required to rotate about its

centre of gravity each elementary mass contained between two planes parallel to the axis o and separated by an infinitesimal distance. The resultant of these couples, F_g, will rotate the body as a whole about its centre of gravity. And second, the force df_o, applied directly at the centre of the elementary mass, produces the acceleration about the axis o. The resultant, Fo, acts at the centre of mass of the body. The moments of these forces will be $df_g \cdot y$ for the elementary mass, or $Fg \cdot R$ for the whole body; and $df_o\ (D - x)$ for the elementary mass about o, or $Fo \cdot D$ for the whole mass.

In more distinctly mathematical language, let D be the distance from the axis o to the centre of gravity of the body, x the distance from the centre of gravity to the elementary mass, y

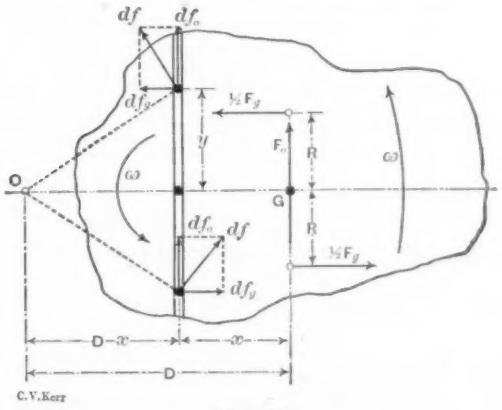


FIG. 59.

the arm of the elementary couple, F the total accelerating force, and I the total amount of inertia of the body. Then, $F = \int df_g + \int df_o = 2\int dm\,\alpha\,y + \int dm\,\alpha\,(D \pm x) = 2\int dm\,\alpha\,y + 2\int dm\,\alpha\,x + \int dm\,\alpha\,D =$

$$M\,\alpha\,\overline{y} + M\,\alpha\,\overline{x} + M\,\alpha\,D, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

in which $\overline{y}$ and $\overline{x}$ are the distances from the gravity axis to the centres of gravity of the body if the whole mass were first projected on the vertical and horizontal planes through the gravity axis and then revolved to the positions GY and GX, as shown in Fig. 60. But $M\,\alpha\,\overline{y}$ and $M\,\alpha\,\overline{x}$ represent two forces at right angles

to each other which may be replaced by their resultant, $M \alpha R$, where $R = \sqrt{\bar{x}^2 + \bar{y}^2}$. Moreover, $M \alpha R$ is the minimum force to produce the required acceleration about the gravity axis, while $M \alpha D$ is the force required to accelerate the whole mass about o if concentrated at the centre of gravity, G. Hence, equation (2) may be written :

$$F = Fg + Fo = M \alpha R + M \alpha D \quad . \quad . \quad . \quad (3)$$

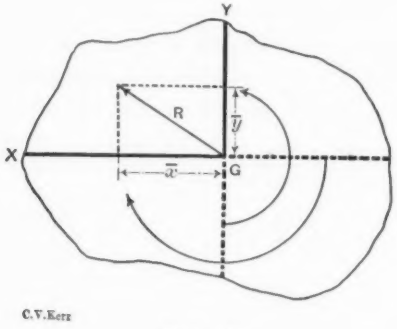


FIG. 60.

Now the moments of the accelerating forces will be $I = \int df_g \cdot y + \int df_o (D \pm x) = \int dm \alpha y^2 + \int dm \alpha D^2 \pm 2 \int dm \alpha Dx + \int dm \alpha x^2 = \int dm \alpha y^2 + \int dm \alpha x^2 + \int dm \alpha D^2$, the term $\pm 2 \int dm \alpha Dx$ dropping out, as it represents two equal and opposite moments. If $D = o$, so that the axis at o coincides with the gravity axis, then

$$I = \int dm \alpha y^2 + \int dm \alpha x^2 = Ig, \quad . \quad . \quad . \quad . \quad (4)$$

in which the term $\int dm \alpha y^2$ measures the moment of the forces rotating about the gravity axis, the whole mass considered projected on a vertical plane, while $\int dm \alpha x^2$ is the corresponding moment for the whole mass projected on a horizontal plane. The general equation of moments about an axis, o, may then be written :

$$I = \int dm \alpha y^2 + \int dm \alpha x^2 + \int dm \alpha D^2 = Ig + M \alpha D^2 \quad (5)$$

If the total moment of inertia be divided by the total accelerating force, the result is the arm, J, of a single force having the same moment; that is,

$$J = \frac{I}{F} = \frac{Ig + M\alpha D^2}{Fg + M\alpha D} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

It has been shown that the moment of inertia of a body may be measured by a single force, I, placed at unit distance from the axis; and, since a body with angular velocity, ω, will move over a space $\frac{1}{2}\omega$ in coming to rest in one second under the action of a constant force, the work done in stopping the rotating body is measured by

$$E = \tfrac{1}{2}\omega I, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

in which E is the kinetic energy in foot-pounds of the rotating body.

Again, since the moment of inertia is measured by a single force at unit distance from the axis, there is evidently a distance at which, if all the mass were concentrated in a single material point, the moment of inertia would be the same. If K be such a distance, then the accelerating force would be $F = M\alpha K$, and its moment would be $I = M\alpha K \,.\, K$. So that, from equation (5),

$$I = Ig + M\alpha D^2 = M\alpha K^2, \quad . \quad . \quad . \quad . \quad (8)$$

the value of K may be obtained. This distance is usually called the "radius of gyration." For the particular case when the axis passes through the centre of gravity, $D = o$ and $M\alpha K^2 = Ig$. This value of K will be known as k; that is,

$$Ig = M\alpha k^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

A *simple pendulum* is by definition a single heavy particle vibrating at the end of a suspending cord without weight. Such a pendulum of weight, W, is represented in Fig. 61. The component of gravity producing acceleration about the axis o is $W \sin\theta$, which is balanced by the inertia force, $M\alpha s$. For a small arc of vibration, the time of oscillation of a simple pendulum is found to be

$$t = \pi\sqrt{\frac{s}{g}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

where t is the time in seconds required for the pendulum to swing from one extreme position to the other.

In distinction from this form, a *compound pendulum* may be any rigid body arranged to oscillate freely about a horizontal axis, as o in Fig. 62. The sum of the gravity forces accelerating the individual particles of the distributed mass will be $W \sin \theta$ acting at the centre of gravity, G. Now, this force produces the rotation of the body about the gravity axis as well as about the axis of suspension, o. Its moment about that axis must, therefore, be equal to the moment of inertia of the body; or, $W \sin$

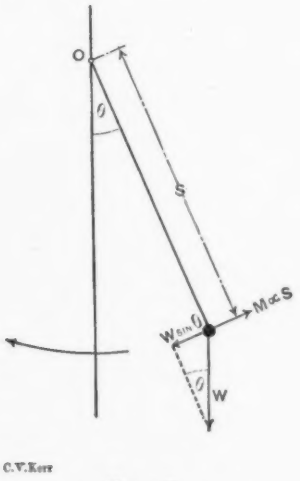


FIG. 61.

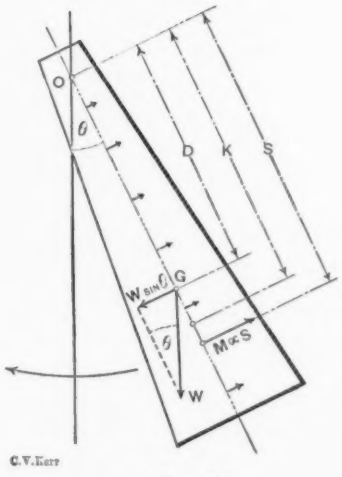


FIG. 62.

$\theta \cdot D = Ig + M \alpha D^2$. For the simple pendulum, where $s = D$, $W \sin \theta = M \alpha s$; and, multiplying by D, $W \sin \theta \cdot D = M \alpha s \cdot D$. Therefore,

$$W \sin \theta \cdot D = M \alpha s \cdot D = Ig + M \alpha D^2, \quad . \quad . \quad (11)$$

from which

$$s = \frac{Ig + M \alpha D^2}{M \alpha D} = \frac{M \alpha k^2 + M \alpha D^2}{M \alpha D} = \frac{k^2 + D^2}{D} \qquad (12)$$

This is the length in feet of a simple pendulum that will oscillate in the same time as the given compound pendulum; and this length is not dependent on mass or acceleration, but simply on form and the distance between axes. It is of interest to note here the limits of s under variation of D. Making s a function

of D in the first and last members of equation (12), and differentiating for a maximum,

$$\frac{ds}{dD} = o = \frac{2D^2 - (k^2 + D^2)}{D},$$

from which $D = k$, and therefore

$$s = 2k \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

If D be made to increase, $\frac{k^2}{D} = o$, finally, and $s = \infty$, or the "centre of oscillation" coincides with the centre of gravity. If D be made to decrease from $D = k$, the result is finally $s = \frac{k^2 + o}{o} = \infty$. The corresponding limits to the time of oscillation will be for the minimum, $t = \pi \sqrt{\frac{s}{g}} = \pi \sqrt{\frac{2k}{g}}$; and for the maximum, $t = \pi \sqrt{\frac{\infty}{g}} = \infty$. Further, from equations (8) and (11), $M \alpha K^2 = Ig + M \alpha D^2 = M \alpha s \cdot D$, from which

$$K = \sqrt{s \cdot D} \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

In other words, the radius of gyration is a mean proportional between the length of the simple pendulum and the distance from the axis of suspension to the centre of gravity.

There is a point in a moving body at which a single force acting in the direction of motion may be applied to stop the body without shock on the axis, or a tendency to disturb the rotation. This point is called the "centre of percussion." Now, the force required to stop a body having a simple motion of translation will be $F = Ma$, applied at the centre of mass. If the body be revolving about an axis, the corresponding force $F = M \alpha D$ applied at the centre of mass will not answer, because it does not affect the rotation about that centre of mass. The angular acceleration and velocity are the same for a revolving body about the gravity axis as about the axis of revolution. Hence the force must be of such a magnitude and so placed as to destroy the two components of the motion of revolution at the same time. Thus in Fig. 63 is represented a body of any form revolving about an axis, o, which may be replaced by the force, Fo, producing rotation about o of the mass considered as

concentrated at the centre of gravity, G, and by the couple, Fg, producing rotation of the distributed mass about the gravity axis. So far as the force $Fo = M\alpha D$ is concerned, it may be balanced by an equal and opposite force, the inertia of the mass concentrated at G; but the couple Fg can theoretically be balanced only by another couple of equal moment—not necessarily of equal force and lever arm. However, by putting in a force, Fo', equal and opposite to the single force, Fo, with a suitable arm, p, a couple may be formed equal to Fg. This latter couple is of such magnitude as to produce an angular acceleration about the gravity axis exactly equal to the acceleration about the axis o effected by Fo. Since couples of equal moment will

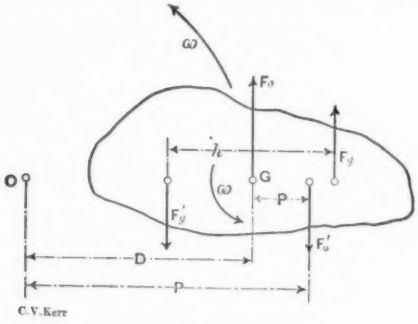


FIG. 63.

do the same work, the value of the lever arm, p, will be determined by the equation of moments $Fo' \cdot p = Fg \cdot h$, from which

$$p = \frac{Fg \cdot h}{Fo'} = \frac{Ig}{M\alpha D} = \frac{M\alpha k^2}{M\alpha D} = \frac{k^2}{D} \quad . \quad . \quad . \quad (15)$$

Consequently the distance from o, at which Fo' is applied to stop the rotation without shock, will be

$$P = p + D = \frac{k^2}{D} + D \quad . \quad . \quad . \quad . \quad . \quad (16)$$

This result is the same in form as the value found for s; and the limiting conditions are the same. Hence it is usual to say that the centre of percussion is identical with the centre of oscillation. But the results from assumed conditions are susceptible of very different meanings. Thus, if D is large, the term $\frac{k^2}{D}$ is a small part of the value of P. For this reason sledge-hammers have a long handle and a concentrated mass, so

that when a blow is struck the centre of percussion coincides closely with the centre of gravity of the head. When $D = \infty$, $P = \infty$, the motion becomes simple translation, and the centre of percussion coincides with the centre of mass. The limiting condition $D = k$ has no special significance beyond making the minimum value of P. But if D is made small, the anomalous result is soon reached of finding the centre of percussion outside the body. Finally, if $D = o$, $P = \infty$. The apparent incongruity of this case simply indicates the limiting condition at which the transfer is made, for example, to a disk or cylinder rotating about its axis, as in Fig. 64. Evidently, no single force

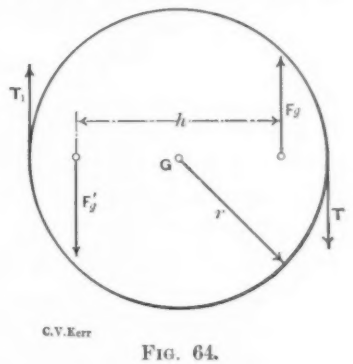


FIG. 64.

can be applied that will stop the rotation of the body without reaction at the axis. But any couple, I, of moment $T \cdot 2r = Fg \cdot h = Ig$ will do so; and that is true even if $T \cdot 2r = o \cdot 2\infty$, which marks the exit from the incongruity.

In the usual sense, the term "centre of percussion" can be used only in cases where it would refer to a single point within the limits of the body. There are, however, in the disk rotating about its gravity axis two points which, if used simultaneously, are true centres of percussion. These points coincide with the points of application of the resultant forces forming the couple Fg, whose moment $Fg \cdot h = Ig$, for this couple imparts or destroys angular acceleration without reaction at the axis. For rotating bodies of such form that the quantities may be calculated, this point may be found by dividing the moment of inertia about the gravity axis by the resultant accelerating force; thus,

$$P = \frac{Ig}{Fg} = \frac{M \alpha k^2}{M \alpha R} = \frac{k^2}{R} \quad . \quad . \quad . \quad . \quad . \quad (17)$$

The fact of coincidence of the centres of percussion and oscillation makes it easy to determine the various quantities for a body of any form experimentally. By determining the time of oscillation of the body used as a compound pendulum, equation (10) gives

$$s = 3.26t^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

in which the constant, 3.26, is the length in feet of the seconds pendulum. Then from equations (9) and (11), having found D by balancing the body on a knife edge, $M \alpha D \cdot s = Ig + M \alpha D^2$, or

$$Ig = M \alpha D (s - D) = M \alpha k^2 \quad . \quad . \quad . \quad . \quad (19)$$

and

$$k = \sqrt{D (s - D)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

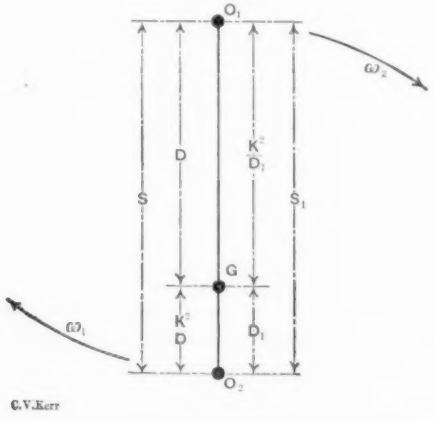


FIG. 65.

Then, knowing Ig, the moment of inertia about any other axis parallel to the axis of oscillation used to determine t may be calculated, since $I = Ig + M \alpha D^2$. And, finally, $P = s$ and $K = \sqrt{k^2 + D^2}$. The time of oscillation does not lead to locating the centre of percussion for a body rotating about the gravity axis; but this is not essential, since any couple of moment equal to Ig will stop or start the rotation without shock or reaction at the axis.

For those conditions under which the centres of oscillation and percussion coincide, those centres are interchangeable with the axis of rotation. Thus in Fig. 65, which represents the centre of gravity of a compound pendulum at G, while the axis

of suspension is first at o_1 and then at o_2, $s = \frac{k^2}{D} + D$, and $s_1 = \frac{k^2}{D_1} + D_1$. But $D_1 = \frac{k^2}{D}$, and, therefore,

$$s_1 = \frac{k^2}{D_1} + D_1 = \frac{k^2}{k^2/D} + \frac{k^2}{D} = D + \frac{k^2}{D} = s \quad . \quad (21)$$

Another point in theory should now be considered. Assume a disk of thickness t and radius r to rotate about an axis G perpendicular to the plane of the disk at its centre, as in Fig. 66. Then the force required to accelerate an element of the mass will be $df = dm \alpha \rho$, and the moment about G will be $df \cdot \rho = dm \alpha \rho^2 = dI$. Then $I = \alpha \int dm \rho^2 = \alpha \int dm (x^2 + y^2) = \alpha \int dm x^2 + \alpha \int dm y^2 = Ix + Iy = 2Ix$, since $x^2 + y^2 = \rho^2$. This is usually called the "polar moment of inertia," and is apparently equal to the "sum of the rectangular moments of inertia about any two axes of the plane figure which intersect at right angles in the given point." But this can be true only of a disk of infinitesimal thickness; for, if the thickness of the disk be increased so that t becomes the length l of a cylinder, the moment of inertia about the axis of the cylinder would be equal to twice the moment about a diameter at the end. But that cannot be true, since the polar moment increases with the length, while the rectangular moment increases with the square of the length. The proposition is more evidently untrue in the case of a sphere, where the moments about all diameters are equal.

A correct relation may, however, be found between the so-called "polar" and "rectangular" moments of inertia. In Fig. 66 the accelerating force df may be replaced by its components $df_x = dm \alpha y$, and $df_y = dm \alpha x$. Then the moments of these forces, both of which cause rotation in the same direction about G, will be $dI = df_x \cdot y + df_y \cdot x = dm \alpha y^2 + dm \alpha x^2 = dm \alpha \rho^2$. But the $\int dm \alpha y^2$ will be the moment of inertia about G if the whole mass were projected on a thin plane, YY^1, through the geometrical axis G, while the $\int dm \alpha x^2$ is the moment about G of the mass projected on the plane XX^1. These planes are indicated by the heavy lines in Fig. 66. They are at right angles to

each other and their intersection is the axis G, about which both rotate in the same direction. In this sense the general equation $I = \int dm \alpha y^2 + \int dm \alpha x^2 + \int dm \alpha D^2$ shows that all moments of inertia are "polar moments."

Thus far, the body considered has been constrained to rotate about the gravity axis in the same manner as about the parallel axis o; so that the factor of acceleration in the general equation $I = Ig + M \alpha D^2 = M \alpha k^2 + M \alpha D^2$ has been common to all the terms. But the mechanical arrangement may be such that the acceleration about the gravity axis will not be the same as about the axis of revolution. For instance, if the body be car-

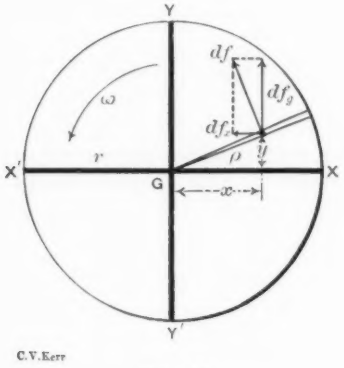


FIG. 66.

ried by a frictionless axis through the centre of gravity parallel to the axis o, then $Ig = o$, and $I = M \alpha D^2$. Or, the arrangement may be such that the acceleration about the gravity axis may be greater or less than the acceleration about o. A notable example is the action of a shaft governor weight during a change of load. For such cases the general equation must be modified so that $I = M \alpha_1 k^2 + M \alpha_2 D^2$, and the various dependent quantities computed accordingly.

In the reduction of formulas to numerals, many mistakes are made on account of the variety of units of measurement in use. It will not be out of place, therefore, to consider here the effect of certain combinations of these units applicable to the equations derived. Thus, since *weight* is the product of the volume of the body by the weight per cubic unit, it will be the same, expressed in pounds, whether dimensions are in feet or inches.

Then $F = M\alpha R = Ma = Weight \times \frac{a}{g}$, in feet $= W \times \frac{a \times 12}{g \times 12}$ $= W \times \frac{a}{g}$, in inches. Hence, the accelerating force is the same in pounds for all dimensions in inches or all in feet. Again, $I = F \cdot P = W \times \frac{a}{g} \times P$, in feet $= W \times \frac{a}{g} \times 12P$, in inches. Hence, if all dimensions are in feet, I is a moment in *foot-pounds*; if in inches, I is in *inch-pounds*.

In illustration of the application of these principles to a particular case, assume a solid cylinder of length l and radius r to be revolving about an axis o distant D from the parallel gravity

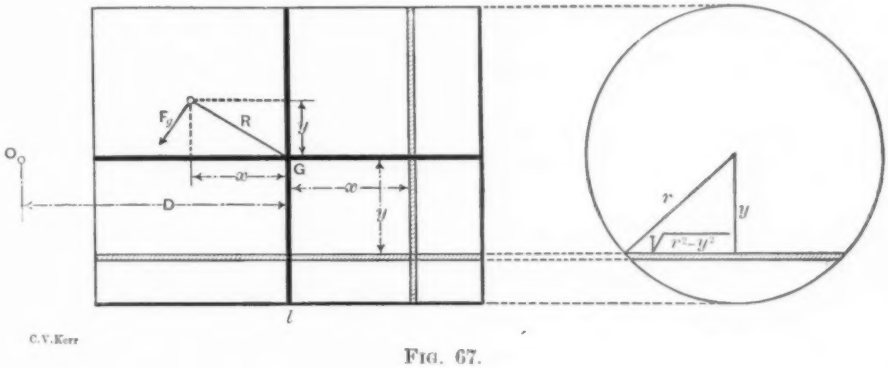


FIG. 67.

axis G, as in Fig. 67. The mass of the cylinder will be $M = \delta\pi r^2 l$, in which δ is the mass per cubic unit. $\bar{x}$ is by inspection $\frac{1}{2}l$. To obtain $\bar{y}$, resort must be had to the equation of moments by weight, $W \cdot \bar{y} = \int dw \cdot y = 2u\int_o^r (r^2 - y^2)^{\frac{1}{2}} y\, dy = \frac{2}{3}ur^3$, in which W is the weight of half a circular section of thickness dx, and u is the weight per cubic unit. Then $\bar{y} = \frac{\int dw \cdot y}{W}$ $= \frac{\frac{2}{3}ur^3}{\frac{1}{2}\pi r^2 u} = \frac{4r}{3\pi}$. The resultant accelerating force will be

$$F = M\alpha R + M\alpha D = M\alpha\sqrt{\bar{x}^2 + \bar{y}^2} + M\alpha D$$

$$= M\alpha\sqrt{(\tfrac{1}{2}l)^2 + \left(\frac{4r}{3\pi}\right)^2} + M\alpha D.$$

The moment of this force, which is equal and opposite to the moment of inertia of the body, will be

$$I = \int dm\,\alpha\,x^2 + \int dm\,\alpha\,y^2 + \int dm\,\alpha\,D^2$$
$$= 2\,\delta\alpha\pi\,r^2 \int_o^{\frac{l}{2}} x^2 dx + 4\,\delta\alpha\,l \int_o^r (r^2 - y^2)^{\frac{1}{2}} y^2 dy + 2\,\delta\alpha\pi\,r^2 D^2 \int_o^{\frac{l}{2}} dx$$
$$= \tfrac{1}{12}\,\delta\alpha\pi\,r^2 l^3 + \tfrac{1}{4}\,\delta\alpha\pi\,r^4 l + \delta\alpha\pi\,r^2 l D^2$$
$$= \tfrac{1}{12}\,M\alpha\,l^2 + \tfrac{1}{4}\,M\alpha\,r^2 + M\alpha\,D^2 = Ig + M\alpha\,D^2.$$

Of these terms, $\frac{1}{12}\,M\alpha\,l^2$ represents the moment of inertia about the gravity axis if the whole mass were projected on the rectangular section of the cylinder made by a plane passing through the axes G and o; $\frac{1}{4}\,M\alpha\,r^2$ represents the moment of inertia of the whole mass projected on the circular section at right angles to the axis of the cylinder; and the sum of these two moments in planes at right angles to each other intersecting in the gravity axis is the true polar moment, Ig, about the gravity axis. And $M\alpha\,D^2$ is the moment about o of the whole mass concentrated on the gravity axis.

The distance from the axis o to the centre of percussion will be

$$P = \frac{Ig}{M\alpha\,D} + D = \frac{\frac{1}{12}\,M\alpha\,l^2 + \frac{1}{4}\,M\alpha\,r^2}{M\alpha\,D} + D = \frac{\frac{1}{12}l^2 + \frac{1}{4}r^2}{D} + D,$$

which in this case coincides with the centre of oscillation. The time of oscillation may be found by the formula $t = \pi\sqrt{\frac{s}{g}}$, in which s is the distance from the axis o to the centre of oscillation, or length of the corresponding simple pendulum.

Assuming the cylinder to be of cast iron 18 inches diameter by 30 inches long, with its centre of gravity 24 inches from the axis o, and that the accelerating force is to increase the speed of rotation three turns per second, then $M = \delta\pi r^2 l = \frac{450}{32.2} \times \pi \times 0.75^2 \times 2.5 = 61.74$ pounds. $\alpha = 2\pi n = 2 \times \pi \times 3 = 18.85$ feet per second. $\bar{x} = \frac{1}{4}l = 0.625$ foot. $\bar{y} = \frac{4r}{3\pi} = \frac{4 \times 0.75}{3\pi} = 0.318$ foot. $R = \sqrt{\bar{x}^2 + \bar{y}^2} = \sqrt{0.625^2 + 0.318^2} = 0.7014$ foot. $F = M\alpha\,R + M\alpha\,D = 61.74 \times 18.85 \times 0.7014 + 61.74 \times 18.85 \times 2 = 816.3 + 2327.6 = 3143.9$ pounds. $Ig = \frac{1}{12}\,M\alpha\,l^2 + \frac{1}{4}\,M\alpha\,r^2 = \frac{1}{12} \times 61.74 \times 18.85 \times 2.5^2 + \frac{1}{4} \times 61.74 \times 18.85 \times 0.75^2 = 606.15 + 163.66 = 769.8$ foot-pounds. $I = Ig + M\alpha\,D^2 = Ig + 61.74 \times 18.85 \times 2^2 = 769.8 + 4655.2 = 5425.0$ foot-pounds. $s = \frac{Ig}{M\alpha\,D} + D =$

$\frac{769.8}{2327.6} + 2 = 0.33 + 2.00 = 2.33$ feet. $Pg = \frac{Ig}{M \alpha R} = \frac{769.8}{816.3} =$ 0.94 foot. $t = \pi\sqrt{\frac{s}{g}} = \pi\sqrt{\frac{2.33}{32.2}} = 0.847$ second.

For convenience of reference and for the sake of determining what quantities connected with the more commonly used solids would be useful in tabular form, the essential equations are grouped as follows:

The resultant accelerating force $F = M \alpha R + M \alpha D = M \alpha \sqrt{x^2 + y^2} + M \alpha D$.

The moment of inertia about the gravity axis, $Ig = \int dm \alpha x^2 + \int dm \alpha y^2 = M \alpha k^2$; from which $k^2 = \frac{Ig}{M \alpha}$, or the radius of gyration, $k = \sqrt{Ig \div M \alpha}$.

The moment of inertia about any axis parallel to the gravity axis, $I = \int dm \alpha x^2 + \int dm \alpha y^2 + \int dm \alpha D^2 = Ig + M \alpha D^2 = M \alpha (k^2 + D^2) = M \alpha K^2$.

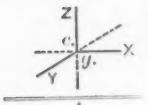
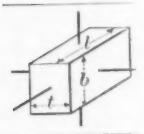
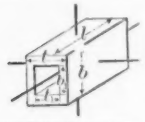
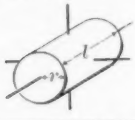
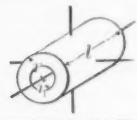
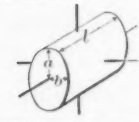
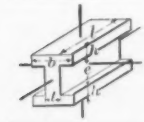
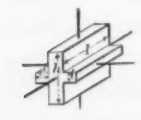
The distance from the axis of suspension to the centre of oscillation, $s = \frac{Ig}{M \alpha D} + D = \frac{k^2}{D} + D$, which is a minimum for $D = k$.

The distance from the axis to the centre of percussion, $P = s$, for rotation about the axis o; but $P = \frac{Ig}{M \alpha R} = \frac{k^2}{R}$ for rotation about the gravity axis.

The radius of gyration about any axis parallel to the gravity axis, $K^2 = k^2 + D^2 = D \cdot s$, or $K = \sqrt{k^2 + D^2} = \sqrt{D \cdot s}$.

The time of oscillation of any solid as a compound pendulum, $t = \pi\sqrt{\frac{s}{g}} = \pi\sqrt{\frac{k^2}{gD} + \frac{D}{g}}$.

Inspection of these equations shows them to be made up of the fundamental quantities, M, $\bar{x}$, $\bar{y}$, Ig, k^2, together with α and D, which are usually given. Hence in the table herewith only those quantities are given for the more useful elementary forms of solids; for by combining these, the desired quantity for a very large number of special cases can readily be obtained.

	Mass of Solid $= \delta V =$ Force in Pounds for Unit Acceleration.	Gravity Axis to Centre of Projected and Revolved Mass. $\left\{ \begin{matrix} \bar{x} \\ \bar{y} \\ \bar{z} \end{matrix} \right.$	Moment of Inertia about the Gravity Axis. $\left\{ \begin{matrix} I_x \\ I_y \\ I_z \end{matrix} \right.$	Sq. of Radius of Gyration $= \frac{I}{Ma}$ $\left\{ \begin{matrix} k_x^2 \\ k_y^2 \\ k_z^2 \end{matrix} \right.$
	$\frac{4}{3}\delta\pi r^3$	In Feet. $\frac{3}{8}r$ $\frac{3}{8}r$ $\frac{3}{8}r$	In Foot-pounds. $\frac{8}{15}\delta a\pi r^5$ $\frac{8}{15}\delta a\pi r^5$ $\frac{8}{15}\delta a\pi r^5$	In Feet. $\frac{2}{5}r^2$ $\frac{2}{5}r^2$ $\frac{2}{5}r^2$
	$\delta l b l$	$\frac{1}{2}l$ $\frac{1}{2}l$ $\frac{1}{2}b$	$\frac{1}{12}Ma(b^2+l^2)$ $\frac{1}{12}Ma(l^2+b^2)$ $\frac{1}{12}Ma(l^2+l^2)$	$\frac{1}{12}(b^2+l^2)$ $\frac{1}{12}(l^2+b^2)$ $\frac{1}{12}(l^2+l^2)$
	$M=M_1-M_2$ $=\delta l b l-\delta l_1 b_1 l$	$\frac{1}{4}\frac{l^2 b-l_1{}^2 b_1}{l b-l_1 b_1}$ $\frac{1}{2}l$ $\frac{1}{4}\frac{l b^2-l_1 b_1{}^2}{l b-l_1 b_1}$	$\frac{M_1 a}{12}(b^2+l^2)-\frac{M_2 a}{12}(b_1{}^2+l^2)$ $\frac{M_1 a}{12}(l^2+b^2)-\frac{M_2 a}{12}(l_1{}^2+b_1{}^2)$ $\frac{M_1 a}{12}(l^2+l^2)-\frac{M_2 a}{12}(l_1{}^2+l^2)$	$\frac{I_x}{Ma}$ $\frac{I_y}{Ma}$ $\frac{I_z}{Ma}$
	$\delta\pi r^2 l$	$4r$ 3π $\frac{1}{2}l$ $4r$ 3π	$\frac{1}{4}Ma r^2+\frac{1}{12}Ma l^2$ $\frac{1}{2}Ma r^2$ $\frac{1}{4}Ma r^2+\frac{1}{12}Ma l^2$	$\frac{1}{4}r^2+\frac{1}{12}l^2$ $\frac{1}{2}r^2$ $\frac{1}{4}r^2+\frac{1}{12}l^2$
	$M=M_1-M_2$ $=\delta\pi l r^2-\delta\pi l r_1{}^2$	$\frac{4}{3\pi}\frac{r^3-r_1{}^3}{r^2-r_1{}^2}$ $\frac{1}{2}l$ $\frac{4}{3\pi}\frac{r^3-r_1{}^3}{r^2-r_1{}^2}$	$\frac{a}{4}(M_1 r^2-M_2 r_1{}^2)+\frac{Ma l^2}{12}$ $\frac{a}{2}(M_1 r^2-M_2 r_1{}^2)$ $\frac{a}{4}(M_1 r^2-M_2 r_1{}^2)+\frac{Ma l^2}{12}$	$\frac{I_x}{Ma}$ $\frac{I_y}{Ma}$ $\frac{I_z}{Ma}$
	$\frac{1}{3}\delta\pi r^2 h$	(c.g. to Vertex $= \frac{3}{4}h$.) $\frac{r}{\pi}$ $\frac{1}{8}\left(\frac{1}{2}\right)^4 h=\frac{81h}{512}$ $\frac{r}{\pi}$	$\frac{3}{20}Ma(r^2+\frac{1}{4}h^2)$ $\frac{3}{10}Ma r^2$ $\frac{3}{20}Ma(r^2+\frac{1}{4}h^2)$	$\frac{3}{20}(r^2+\frac{1}{4}h^2)$ $\frac{3}{10}r^2$ $\frac{3}{20}(r^2+\frac{1}{4}h^2)$
	$\delta\pi a b l$	$4b$ 3π $\frac{1}{2}l$ $\frac{4a}{3\pi}$	$Ma(\frac{1}{4}a^2+\frac{1}{12}l^2)$ $\frac{Ma}{4}(a^2+b^2)$ $Ma(\frac{1}{4}b^2+\frac{1}{12}l^2)$	$\frac{1}{4}a^2+\frac{1}{12}l^2$ $\frac{1}{4}(a^2+b^2)$ $\frac{1}{4}b^2+\frac{1}{12}l^2$
	$M=M_1+M_2$ $=2\delta b h l+\delta l e l$	$\frac{\frac{1}{2}l^2 e+\frac{1}{2}b^2 h}{l e+b h}$ $\frac{1}{2}l$ $\frac{\frac{1}{2}l e^2+\frac{1}{2}b h(e+h)}{l e+b h}$	$\frac{1}{12}M_1 a(h^2+l^2)+\frac{1}{4}M_1 a(e+h)^2+\frac{1}{12}M_2 a(e^2+l^2)$ $\frac{1}{12}M_1 a(h^2+h^2)+\frac{1}{4}M_1 a(e+h)^2+\frac{1}{12}M_2 a(l^2+e^2)$ $\frac{1}{12}M_1 a(h^2+l^2)+\frac{1}{12}M_2 a(l^2+e^2)$	$\frac{I_x}{Ma}$ $\frac{I_y}{Ma}$ $\frac{I_z}{Ma}$
	$M=M_1+M_2$ $=\delta b h l+2\delta l e l$	$\frac{\frac{1}{2}h^2 h+\frac{1}{2}l e(e+b)}{b h+2l e}$ $\frac{1}{2}l$ $\frac{\frac{1}{2}l^2 e+\frac{1}{2}b h^2}{l e+\frac{1}{2}b h}$	$\frac{1}{12}M_1 a(h^2+l^2)+\frac{1}{12}M_2 a(l^2+l^2)$ $\frac{1}{12}M_1 a(h^2+h^2)+\frac{1}{12}M_2 a(l^2+e^2)+\frac{1}{4}M_2 a(e+b)^2$ $\frac{1}{12}M_1 a(h^2+l^2)+\frac{1}{12}M_2 a(e^2+l^2)+\frac{1}{4}M_2 a(e+b)^2$	$\frac{I_x}{Ma}$ $\frac{I_y}{Ma}$ $\frac{I_z}{Ma}$

It would add to the completeness of this paper to discuss at this point the so called "ellipsoid of inertia"; but it is difficult to present that phase of the subject clearly without being tedious, and as the results are seldom, if ever, useful in practice the discussion is omitted.

Solids of irregular shape or combinations of solids may not yield readily, if at all, to analytical methods of calculation. To meet such cases, as well as those yielding to calculation, the following graphical method is well adapted. In order to secure verification and a comparison of results as to accuracy, the cylinder is resumed as an example for which the various quantities have already been computed. This cylinder is shown in Fig. 68, drawn to one-eighth scale and divided, first, by planes parallel to the bases into ten circular disks whose centres of gravity are at 1, 2, 3, . . . 10; and, second, by planes parallel to the axis of the cylinder into slabs whose centres of gravity are at 1′, 2′, 3′, . . . 10′. As cast iron weighs about 450 pounds per cubic foot, the mass of each disk is $\delta \pi r^2 l = \frac{450 \times 254.47 \times 3}{1728 \times 32.2} = 6.18$ pounds. Assuming the convenient scale of 20 pounds per inch, or 240 per foot, lay off on the "load line" sl the distances 1, 2, 3, . . . 10 each equal to 6.18 pounds. From any point, p, in line with the other base of the cylinder, draw ps and pl. From the centre (1) drop a perpendicular from O_1 O_5, to intersect ps at a. Draw a parallel to ph from a to intersect a perpendicular from centre (2) at b; and proceed thus until the polygon $acfg$ is formed, the last chord, fm, being parallel to lp. Produce mf to intersect ps at c; draw gcG and extend ab to e. Now the triangles ace and psh are similar, and $x : \overline{ce} = H_x : \overline{sh}$, or $x \cdot \overline{sh} = H_x \cdot \overline{ce}$. In the general equation for the moment of inertia, $I = \int dm \, \alpha \, x^2 + \int dm \, \alpha \, y^2 + \int dm \, \alpha \, D^2$, the dm is represented here by $\overline{sh}$. Hence the proportion, with the factor α introduced, becomes $dm \, \alpha \, x = \alpha \, H_x \cdot \overline{ce}$. But $dm \, \alpha \, x$ is the accelerating force for the mass whose centre is at (1); and, consequently, the accelerating force required for the masses 1, 2, . . . 5 will be $\alpha \, H_x \cdot \overline{cg}$. Similarly, for the masses 6, 7, . . . 10, the force will be $\alpha_x H \cdot \overline{cg}$, which is equal and opposite to the force required to accelerate the masses 1, 2, . . . 5, although causing rotation in the same direction. The point c is

therefore on a line cG through the centre of gravity of the body; and the entire accelerating force about the gravity axis will be $Fg = 2\alpha\ H_x \cdot \overline{cg}$.

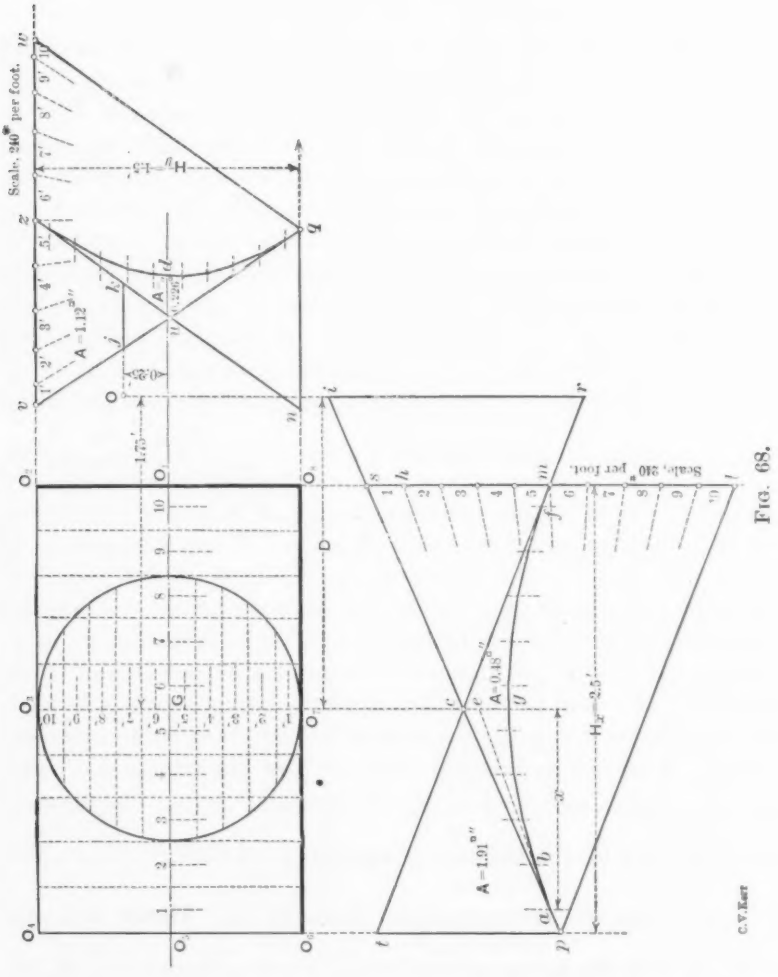


Fig. 68.

The moment of the accelerating force for the mass (1) will be $dm\,\alpha\,x^2 = \alpha\ H_x \cdot \overline{ce} \cdot x = 2\,\alpha\ H_x \cdot \frac{1}{2}\overline{ce} \cdot x = 2\alpha\ H_x \times$ area triangle $\overline{ace}$; and the $\int dm\,\alpha\,x^2 = 2\alpha\ H_x \times$ area $\overline{acfg} = 2\alpha\ H_x\ A_x$. Similarly, find the polygon $\overline{quzd}$. Then two lines at right angles to each other through c and u will intersect in the centre of mass, G, of

the cylinder; and the $\int dm \alpha y^2 = 2\alpha H_y \times$ area $\overline{quzd} = 2\alpha H_y A_y$. Then,

$$Ig = \int dm \alpha x^2 + \int dm \alpha y^2 = M \alpha k^2 = 2\alpha H_x A_x + 2\alpha H_y A_y \quad (22)$$

The numerical result is to be a moment in foot-pounds so that H must be expressed in feet from the scale of the drawing; while A_x or A_y, representing areas, will be affected in one dimension by the scale of the drawing and in the other by the scale of the load line $\overline{sl}$ or vw. As these areas will usually be determined by the planimeter in square inches, the area found must be divided by 144 to reduce to square feet, then multiplied by 8 if the scale of the drawing is one-eighth, and by the scale of the load line in pounds per foot.

Having assumed an axis through any point o parallel to the gravity axis, the triangles pls and cir are similar, so that $H : \overline{sl} = D : \overline{ir}$, or $\overline{sl} \cdot D = H \cdot \overline{ir}$. But $\overline{sl}$ represents the entire mass, M, of the cylinder. Introducing α, the proportion gives $M \alpha D_x = \alpha H_x \cdot \overline{ir}$, in which H_x is to be measured in feet and $\overline{ir}$ is a force in pounds. Similarly, $M \alpha D_y = \alpha H_y \cdot \overline{jk}$. Then, finding the resultant of the two forces about the gravity axis and of the two about the given parallel axis through v, the result is

$$F = Fg + Fo \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

The moment of the accelerating force $M \alpha D_x = \alpha H_x \cdot \overline{ir}$ will be $M \alpha D_x \cdot D_x = \alpha H_x \cdot \overline{ir} \cdot D_x = 2\alpha H_x \times \frac{1}{2}\overline{ir} \times D_x = 2\alpha H_x \times$ area triangle $\overline{cir} = 2\alpha H_x A_x^1$; and $M \alpha D_y^2 = 2\alpha H_y A_y^1$. Since $D_x^2 + D_y^2 = D^2$, then $M \alpha D_x^2 + M \alpha D_y^2 = M \alpha D^2 = 2\alpha H_x A_x^1 + 2\alpha H_y A_y^1$, and the total moment about o measured in foot-pounds will be

$$\begin{aligned} Io &= 2\alpha H_x (A_x + A_x^1) + 2\alpha H_y (A_y + A_y^1) \\ &= 2\alpha H_x \times \text{area } \overline{pcirg} + 2\alpha H_y \times \text{area } \overline{qjkzd}. \end{aligned} \quad (24)$$

The triangles cir and cpt are similar, so that the accelerating forces and the moment of inertia will be the same for equal positive and negative values of the ordinates to o, the area $acfg$ being common. And inspection of the diagrams will show that by proper combination of data from the polygons $acfg$ and $quzd$, and the triangles pct, csm, unq, vzu, the accelerating forces and the moments of inertia of the cylinder about the nine parallel axes, G, O_1, O_2, . . . O_8, may be obtained.

The results from the graphical solution may, of course, be used as in the analytical method for the determination of the centres of percussion or oscillation and the radii of gyration.

In comparing the results from the analytical and graphical methods, the measurements from the diagrams were taken while in fine pencil lines. The location of the centre of gravity of the solid is known in this case. The point c came out exactly right, the point u nearly so. From the equations for moment of inertia, the area *ptcmg* should be equal to four times the area *acfg*. This is correct within 0.01 square inch, the limit of the planimeter used. Likewise $5 \times$ area $quzd = 1.13$ square inches, instead of 1.12 from the polygon *quvzd*.

The analytical method gives for the accelerating force about the gravity axis, $F_g = M \alpha R = 816.3$ pounds; while by the graphical, $F_x = 2\alpha H_x \cdot \overline{cg} = 2 \times 18.85 \times 2.5 \times 0.38 \times 20 = 716.3$ pounds, and $F_y = 2\alpha H_y \cdot \overline{ud} = 2 \times 18.85 \times 1.5 \times 0.34 \times 20 = 384.5$ pounds, the resultant being $F_g = \sqrt{F_x^2 + F_y^2} = 813$ pounds. The difference is less than 0.4 of 1 per cent. of either. By the analytical method, $I_g = 769.8$ foot-pounds; by the graphical, $\int dm \, \alpha \, x^2 = 2\alpha H_x \cdot$ area $\overline{acfg} = 2 \times 18.85 \times 2.5 \times \frac{0.48}{144} \times 8 \times 240 = 603.2$ foot-pounds, and $\int dm \, \alpha \, y^2 = 2\alpha H_y \cdot$ area $\overline{quzd} = 2 \times 18.85 \times 1.5 \times \frac{0.226}{144} \times 8 \times 240 = 170.4$ foot-pounds, the sum of moments being $Ig = 603.2 + 170.4 = 773.6$ foot-pounds. The difference is a half of one per cent. of either.

Again, the moment of inertia about O_1 in Fig. 68 is $I = Ig + M \alpha D^2 = Ig + 61.74 \times 18.85 \, (1.25^2 + 0.75^2) = 769.8 + 2473.1 = 3242.9$ foot-pounds. From the diagrams, $I = 2\alpha H_x \cdot$ area $\overline{mctpg} + 2\alpha H_y \cdot$ area $\overline{quvzd} = 2 \times 18.85 \times 2.5 \times \frac{1.91}{44} \times 8 \times 240 + 2 \times 18.85 \times 1.5 \times \frac{1.12}{144} \times 8 \times 240 = 2400.2 + 844.5 = 3244.7$ foot-pounds. The difference here is but 0.05 of one per cent.

Finally, for an axis parallel to the gravity axis passing through any point, as o, whose coördinates referred to G are $x = 1.75$ feet and $y = 0.25$ foot, $I = Ig + M \alpha D^2 = Ig + 61.74 \times 18.85 \, (1.75^2 + 0.25^2) = 769.8 + 3636.9 = 4406.7$ foot-pounds. While $I = Ig + 2\alpha H_x \cdot$ area $\overline{eir} + 2\alpha H_y \cdot$ area $\overline{ujk} = 773.6 + 2 \times 18.85 \times 2.5$

$\times \frac{2.62 \times 2.15}{2 \times 144} \times 8 \times 240 + 2 \times 18.85 \times 1.5 \times \frac{0.39 \times 0.56}{2 \times 144} \times 8 \times 240 = 773.6 + 3539.4 + 82.3 = 4395.3$ foot-pounds. In this, and all other cases in fact, the moment about o may be found without first knowing the moment about G by making $I = 2\alpha\, H_x \cdot$ area *peirg* $+ 2\epsilon\, H_y \cdot$ area *qujkzd*, since the areas used added together equal these. The difference here is 0.26 of one per cent.

These results are in practical agreement. The purpose in this comparison has been to show the correctness of the methods used, in part the means available to check results, and the degree of accuracy to be expected with ordinary care in calculation and drawing. So far as the graphical method is concerned, the larger the scale used within convenient limits, the greater the accuracy; and care is essential always.

In conclusion, the salient features of this form of the theory are considered to be:

(1) That *mass* may be used as a measure of force causing unit acceleration in unit time.

(2) That *momentum* is not merely a "quantity of motion," but a measure of the inertia developed in a body.

(3) That the *moment of inertia* of a body depends upon *acceleration* as well as upon *mass*, and is expressed as the moment of a force in inch or foot pounds.

(4) That the centres of oscillation and of percussion are not always identical.

(5) That in this, the concrete form, the moment of inertia is a question of dynamics, as opposed to the moment of resistance, which is a question of statics.

(6) That the analytical and graphical methods evolved are either singly or in combination capable of effecting a solution of any useful problem.

(7) That the concrete form of theory permits the designation of a numerical result in rational terms.

DISCUSSION.

Mr. Wm. Kent.—The object of the author, as it appears in his paper of 1894 as well as in the present paper, is to give students some new ideas of the moment of inertia, and to help them form some mental conception of what it is. It may be allowable in discussion to make a restatement, in possibly a new form, of old

ideas on the subject. Let us consider first what Rankine calls the geometrical moment of inertia of the cross-section of a beam. Suppose a plank 12 inches wide to be sawed through, and the two pieces, separated a short distance, to be joined in the middle by a hinge or flexure pivot at O, as in Fig. 69. Place a thin strip of elastic material, like rubber, in the space on one side of the pivot. Let the elastic compressive resistance of the rubber be such that f pounds will compress a lineal inch of the strip .01 inch; $6f$ pounds .06 inch, etc. Apply a force F, acting at a lever arm of 10 inches from the pivot, so as to produce a compression of .06 inch at the extreme edge, or 6 inches from the pivot, and a compression of .01 inch 1 inch from the pivot. The force exerted on the rubber at the extremity of the radius r will be $6f$ pounds per lineal inch, and f pounds per lineal inch 1 inch from the pivot. Conceive the strip to be cut into a great number of short sections. The outer section will resist compression at the rate of 6 pounds per inch, and the innermost section, adjoining the pivot, will not be compressed at all, and its resistance will be O. The sum of the resistances of all the sections, or of the whole strip, may be represented graphically by the area of a triangle (Fig. 70) whose base is $6f$ pounds and whose altitude is r or 6 inches, the area being 18 f pounds. Each small section of the strip resists the force F, not only in proportion to its compression, which causes a resistance varying from $6f$ to 0 pound, but also in proportion to the lever arm, at which each particular section acts; that is, the outermost section has a lever arm of 6 inches and a compression of .06 inch, and has a moment of resistance per lineal inch of $6f \times 6 = 36f$ inch-pounds. The sum of the moments of all the strips may be represented by the volume of a pyramid (Fig. 71) whose altitude is r, or 6 inches, the base one of whose sides is 6 inches, and the adjoining side of the base $6f$ pounds. This volume, according to mensuration, is $\frac{1}{3}$ of $36 \times 6 = 72$ square inches $\times$ f pounds. At the distance of 1 inch from the pivot, the section has a resistance of f pounds per inch, and its lever arm is 1 inch, the moment being f inch-pounds per lineal inch. At the distance r or 6 inches it is r^2 or 36 inch-pounds per lineal inch. The moment, therefore, varies as the square of x, which is any part of r. We may make the summation of the moments by the calculus, thus:

Total moment of resistance $= \int_0^r fx^2 dx = \frac{1}{3}r^3 f$, or $72f$. In the above calculations the breadth of the strip has been considered

as unity. If the breadth be b the total moment will be $\int_o^r f\, bx^2\, dx = \frac{1}{3}\, br^3 f$. The calculations also assume that the strip is rectangular, which makes the graphical solution, by the device of the pyramid, possible. The method by the calculus is general for any shape of strip, but the integration is more or less difficult, according to the shape.

Now assume that we have a similar rubber strip on the other side of the pivot, resisting tension (being properly fastened or glued), in the same way that the first strip resists compression, all conditions of elasticity, etc., being the same, and let the applied force F be doubled. The total moment of resistance will then be

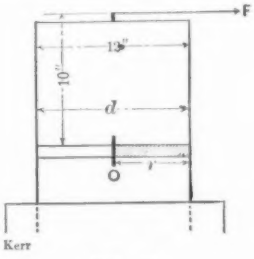


FIG. 69.

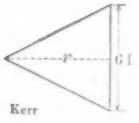


FIG. 70.

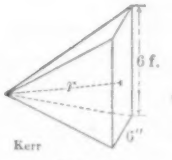


FIG. 71.

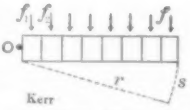


FIG. 72.

$\frac{2}{3}\, br^3 f$. If we substitute d, or the depth of the beam for $2r$ or d^3 for $8\, r$, we obtain total moment $= \frac{1}{12}\, bd^3 f$.

In both expressions $\frac{1}{3}\, br^3 f$ and $\frac{1}{12}\, bd^3 f$, we see that the moment is composed of two factors, one f, the rate of the force at unit distance, and the other $\frac{1}{3}\, br^3$ or $\frac{1}{12}\, bd^3$, a function of the magnitude and shape of the area compressed. To this function, which is generally $\int_o^r bx^2\, dx$, or $2\int_o^r bx^2\, dx$ or $\Sigma\, ar^2$, in which latter expression a is a small elementary area, r the distance of each elementary area from the axis of rotation, and Σ the sign of summation, has been given the name "moment of inertia." The name may not be a good one, for the function is not a moment and it is not inertia, but it has so long been incorporated into the literature of engineering that it is not likely to be displaced. It might be called area function or "area modulus." It might be called "section modulus," but that term has already been appropriated by another function, viz.: this so-called moment of inertia divided by the distance of the extreme fibre from the axis of rotation. The "section modulus" is

improperly called "moment of resistance" by some authors. What is this "moment of inertia"? It is simply a function, an algebraic expression. It is Σar^2, or in the case of a rectangular area rotated about one edge, $\frac{1}{3} br^3$, or in the case of a rectangular area rotated about an axis in the middle, parallel to one edge, $\frac{1}{12} bd^3$. It is a very convenient algebraic expression, for use in computing the resistance of beams and columns, and it may be used just as well without a mental conception of it as with one. It is, moreover, a function of the magnitude and shape of an area, and of the position of an axis of rotation of that area, and of nothing else. It has nothing to do with either force, inertia, or with anything else but an area.

Moment of Inertia of a Rotating Body.

Conceive the body shown pivoted at O, Fig. 72, divided into a number of separate sections, each one of which is acted on by a separate force f^1, f^2, etc., for a short period of time, the magnitude of each force being such as to cause each corresponding section to acquire the same angular velocity at the same period of time, or to give it the same angular acceleration. Suppose the force f^1, acting on the small section whose mass is m, at the extreme end of the radius r, gives it a lineal motion s at the end of a small period of time, causing it to acquire a velocity $v = \omega r$, ω being the angular velocity. The work done by the force is $fs = \frac{1}{2} mv^2 = \frac{1}{2} m\omega^2 r^2$. The small sections all having the same mass m, the forces impelling them will vary with the acceleration of the several masses from the maximum force f down to 0. The total work done in accelerating the whole body will be $\Sigma fs = \Sigma \frac{1}{2} m\ \omega^2 r^2$. This expression is divisible into two parts, viz.: $\frac{1}{2}\omega^2$ and Σmr^2. The first is a function of the angular acceleration, and hence of the acting force. The second is a function of the mass of each elementary particle, which depends on four elementary factors, its length, its breadth, its depth, and its specific heaviness or density, and also on the geometrical shape and magnitude of the body. This expression Σmr^2 is also called *moment of inertia*, but it is a more complex function than the geometrical moment of inertia of the cross section of a beam. It is also called the moment of inertia of the mass of a body, or $\Sigma \frac{w}{g} r^2 = \Sigma\, mr^2$, to distinguish it from the moment of inertia of the weight of a body, Σwr^2, which is

more commonly used in mechanical engineering, as stated by Rankine (*Machinery and Millwork*, p. 459). Rankine also defines the sum Σwr^2 as "the weight of a body which, if concentrated at the distance unity from the axis of rotation, would require the same work to produce a given increase of angular velocity which the actual body requires.

The subject of the different moments of inertia seems to be treated with sufficient clearness and to a sufficient extent by the standard authorities, such as Weisbach and Rankine, and a student who has had the requisite preliminary training in mathematics should have no difficulty in understanding it after a little explanation from his instructor. There may be a need of rewriting Rankine in more elementary form, so as to make it more comprehensible by the average student, but what is not needed is any such mathematical discussion of it as Professor Kerr has given us in his two papers. I do not see that they have improved in the slightest degree on the treatment of the text books nor brought out any new facts of importance.

In regard to Professor Kerr's conclusions I must take exception to the first three, leaving to others the discussion of the other four. His first conclusion, that mass may be used as a measure of force, tends simply to a confusion of ideas. This statement may be derived from the equation $ft = mv$ if each of the qualities v and t be taken as unity, but we might as well derive from it that time may be used as a measure of mass: $t = \frac{mv}{f}$, when v and f are both taken as unity. It may be curious, but it is not useful.

2. Momentum is merely the so-called "quantity of motion," and it is strictly the product of m and v, and nothing else. The term "inertia developed in a body" is, I think, new, but it is vicious. We might as well speak of area developed in a plane surface or of weight developed in a body.

3. The moment of inertia does not depend upon acceleration. It is Σwr^2, w being a function of the heaviness and of the dimensions of the body, and r a function of its shape. There is in the expression nothing which corresponds to force, velocity, or acceleration.

The moment of inertia treated of by the author seems to be an entirely different thing from what is called moment of inertia in the text-books. My discussion has relation solely to the latter. If his moment of inertia is a different thing, he should give it a different name, say moment of resistance, referring to the resistance

of a beam to bending, or work of angular acceleration, referring to rotating masses.

Mr. Kingsbury.—I wish to say, with reference to clause No. 4, "That the centres of oscillation and of percussion are not always identical," that I have never met that statement before. On the contrary, it seems to be proven with mathematical exactness that the centre of oscillation and the centre of percussion do coincide in all cases.

*Prof. C. V. Kerr.**—So far as names are concerned, there is no need of new ones. I am simply trying to put more life into a few of the old ones, to invest the "moment of inertia" with its true and full meaning, and to distinguish it from the old "moment of resistance," which is purely a question of static stress. But the fact that, thus treated, both moments reduce to foot-pounds is an illustration of the unity that may appear in heterogeneity.

In his *Pocket Book for Mechanical Engineers*, Mr. Kent is supposed to state the accepted definitions in mechanics. Thus on p. 415: "A force should always mean the pull, pressure, rub, attraction or repulsion of one body upon another, and always implies the existence of a simultaneous, equal, and opposite force exerted by that body on the first body, *i. e.*, the reaction." "The unit of force used in engineering, by English writers, is the pound avoirdupois." "Inertia is that property of a body by virtue of which it tends to continue in the state of rest or motion in which it may be placed until acted on by some force." Under these definitions, what supplies the forces acting during impact in the case of two free moving bodies, unless it is inertia? In fact, *is inertia merely a measure of the forces called action and reaction, or is it a force?* And yet I have not gone farther than to use inertia as the *measure* of force.

Again, p. 427: "Force equals the product of mass by the acceleration, or $f = ma$." "The resistance to acceleration is the same as the force applied to produce acceleration." Now, if the acceleration, $a = 1$, then $f = m$, and *mass*, $m = \frac{w}{g}$, becomes the measure of the force in pounds required to impart unit acceleration. That is the property of matter which is constant throughout the known universe.

Finally, on p. 428: "Momentum, or quantity of motion in a

* Author's closure, under the Rules.

body, is the product of the mass by the velocity at any instant $= mv = \frac{w}{g} v$. . . . Since $ft = mv$, if $t = 1$ second, $mv = f$ whence momentum might be defined as numerically equivalent to the number of pounds of force that will stop a moving body in 1 second, or the number of pounds of force which, acting during 1 second, will give it the given velocity." This last sentence contains an idea which was presented in my first paper on the moment of inertia, read before the Society in 1894, some months before the issue of the first edition of Mr. Kent's book. However, he now reaffirms, with Newton, that momentum is merely the so-called "quantity of motion"—strictly the product mv, and nothing else.

My use of momentum as a measure of the inertia developed in a body is better compared to the power developed in a given steam engine which depends upon speed and steam pressure than to the area developed in a plane surface, or to weight developed in a body, as Mr. Kent compares it. But area is merely a definite portion of the plane itself, and is variable between zero and the limit of the plane, which is infinity. And according to Mr. Kent, p. 427: "Weight varies according to the variation in the force of gravity at different places," and consequently a man's weight may vary from zero in midspace to 150 pounds on the earth's surface or even to infinity on some mighty planet or sun in the universe. If $f = ma$, then in a given body the resistance to acceleration—another name for inertia—may vary with the acceleration between zero and infinity. Some other word than "develop" might more neatly express the fact, but I do not yet agree that it is *vicious* to speak of inertia developed in a body.

In a recent paper read before the American Association for the Advancement of Science, Mr. Kent used the equation $f = \frac{mv}{t}$ as a basis for the definition that "unit time is that time in which unit mass will acquire unit velocity when acted on by unit force." If time is to be made a dependent variable, then this definition is not a bad statement of fact. In this discussion, the effect of this adroit change is apparently to cast discredit on a sound statement.

I am indebted to Mr. Kent for the exhibit he has made of the modern form of the old way of deducing the various moments of inertia, and especially so as he has charged one, at least, of my

conclusions with tending to a confusion of ideas. He shows us no less than three variations, Σar^2, Σwr^2, and Σmr^2, all different so far as form is concerned, and so far as meaning can be put into them; and yet they are all called "moment of inertia." He says of the first, Σar^2, "The name may not be a good one, for the function is not a moment, and it is not inertia." That was well said. And what shall we call this moment of inertia when reduced to figures? It is amusing at first and then alarming to note the straits to which even our best writers on engineering mechanics are driven to find a suitable name, for we all feel a natural craving for some kind of name. Thus, Professor Church (*Mechanics of Engineering*, p. 454) states the moment of inertia of a plane figure in terms of "biquadratic inches"; and Professor Merriman (*Mechanics of Materials*, p. 57) gives the moment of inertia of the cross section of an I beam in term of "inches^4," or inches to the fourth power. Another illustration of a somewhat different nature of the difficulty growing out of the use of that convenient abstraction, Σar^2, is found in Professor Hoskins' (*Graphic Statics*, p. 159) definition of "the moment of inertia of a force"; but he frankly confesses that "It is only in the case of masses that the term 'moment of inertia' is really appropriate."

In my paper on the "Moment of Resistance" (A. S. M. E., vol. xviii.) I show how to find the resisting moment of a beam directly from the cross section; and, in the case of a rectangular beam, it is $R = \frac{1}{6}\, sbd^2$, in which s is the stress in pounds per square inch in the extreme fibre, b the breadth, and d the depth. The old and accepted method makes $R = \dfrac{s\,\Sigma ar^2}{e} = \dfrac{s\mathrm{I}}{e} = \dfrac{s}{\frac{1}{2}d} \cdot \dfrac{bd^3}{12} = \frac{1}{6}\, sbd^2$. The result is the same in form, but the labor must always be greater, since R can be tabulated as readily as Σar^2. And, by treating the question purely as a question in statics, the numerical result becomes at once the moment of a force in pounds at unit distance from the axis.

For a distinction between Σmr^2 and Σwr^2 affording food for thought I would cite Rankine's *Applied Mechanics*, p. 514; and he further says that when the term is used in the strict sense, the "moment of inertia will be expressed by the product of a certain number of *pounds avoirdupois* into the square of a certain number of *feet*." But what shall the numerical result be called?

Weisbach (*Theoretical Mechanics*, pp. 576–578) uses acceleration in his analysis, but omits it in the result, and says the moment of inertia of a mass m is $w = mr^2$. He speaks incidentally in an example of a "moment of inertia of 50 foot-pounds." In the method adopted in this paper, the rotation of a rigid body about an axis is kept in mind throughout, while *mass* and *acceleration* are not only *used*, but *retained*. Consequently, the numerical result is shown to be the moment of a force in pounds at one foot from the axis of rotation, which is equivalent to the actual system of forces required to impart to the given body the desired acceleration. Again, the value of I can be as readily tabulated as Σmr^2 or Σwr^2, and in use part of the mental labor of substitution is saved.

Moreover, where is the confusion of ideas charged? Treating these questions as I have, shows that the resisting moment of a beam under a static load and the moment of inertia of a rotating body both reduce to a moment measured in foot-pounds or in inch-pounds. Over against this you must place, on behalf of the accepted theories, biquadratic inches, inches^4, pounds avoirdupois into the square of a certain number of feet, and so on.

In answer to the objection raised by Mr. Kingsbury, the centres of percussion and oscillation apparently do coincide as shown by the equation $S = P = \frac{k^2}{D} + D$. But suppose that a circular disk is rotated about an axis parallel to the axis of its cylinder, and that the two axes are caused to approach and finally to coincide, the values of S and P remain equal and increase to infinity as the axes coincide. At this point, as an oscillating body, the disk will cease to oscillate; but as a rotating body, the disk may continue to rotate with the initial angular velocity. Rankine says (*Applied Mechanics*, p. 522): "No centre of percussion exists when the axis traverses the centre of gravity of the body." But the body rotates; and how shall it be stopped without shock on the axis as before? The analysis illustrated by Fig. 63 of this paper shows that the force F_o', applied at the centre of percussion, constitutes with F_o a couple whose moment is always equal and opposite to that of F_g, F_g'. This latter couple still exists when the axis coincides, and may be opposed by another couple, T, T', of equal moment as in Fig. 64. The true centres of percussion in this case are the centre of percussion of the wedge-shaped mass between two radial planes, subtended by a small angle. The fact

seems to be that a rotating body, if stopped without shock at the axis, is always stopped by a couple, and not by a single force. Hence, when a body rotates about the gravity axis, the centre of oscillation is at infinity, while the centre of percussion is within the body.

So far as new definitions of terms are concerned, enough has been said. As a whole, the method used is original with me, and this paper on the moment of inertia is the final expression of an idea that has gradually developed during the past ten years. The graphical method evolved for solving the general equation, $I = \int dm \, \alpha \, x^2 + \int dm \, \alpha \, y^2 + \int dm \alpha \, D^2$, has been a source of genuine pleasure to me on account of its simplicity and comprehensiveness. It constitutes, moreover, a long step toward a possible science of graphical dynamics. Much time and labor have been spent in my efforts to put life into these concepts of mechanics, but I believe the results are stable and useful.

DCCC.*

THE BURSTING OF SMALL CAST-IRON FLY-WHEELS.

BY CHAS. H. BENJAMIN, CLEVELAND, O.

(Member of the Society.)

Of late years the failures of large fly-wheels have become alarmingly common. Every month brings its record of one or more disasters of this sort, some of them entailing loss of life and serious destruction of property. It is not the purpose of this paper to discuss the causes of such accidents, further than to notice the fact that the high belt speeds and close regulation required in electric plants have been indirectly responsible. Many of the fly-wheels have failed on account of excessive speed due to disarrangement of the governor and consequent racing of the engine. In some instances it has been difficult to determine the cause on account of the destruction and excitement at the moment. In no small number of instances, however, the wheels have burst at speeds but slightly above the normal and when the factor of safety was apparently ample.

Mr. James Stanwood, of this Society, was the first to point out the condition of stress existing in a fly-wheel rim and to show that the bending due to centrifugal force might reduce very materially the bursting speed.† This subject was further developed by Professor Lanza, and the probable amount of stress due to bending was indicated as well as its effect upon rim joints.‡ It occurred to the writer that a series of experiments on small cast-iron wheels might throw some light on the causes of failure and lead to more rational formulas for design.

The quality of the metal in a small wheel is better than in a large, and the stresses due to uneven cooling are much less. The linear speed of rim at which a large wheel will burst will therefore be less than that obtained by experiments on small wheels.

* Presented at the New York meeting (November, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

† *Transactions* A. S. M. E., vol. xiv., p. 251.

‡ *Transactions* A. S. M. E., vol. xvi., p. 208.

The experiments about to be described were conducted under the immediate direction of the writer at the laboratories of Case School of Applied Science, and he was present at the bursting of nearly every wheel. Acknowledgment should be made of the services of the students who carried out the experiments, Messrs. Bishop and French of the class of '97 and the Messrs. Emrich, of the class of '98. Without their intelligent and faithful assistance the work could not have been done.

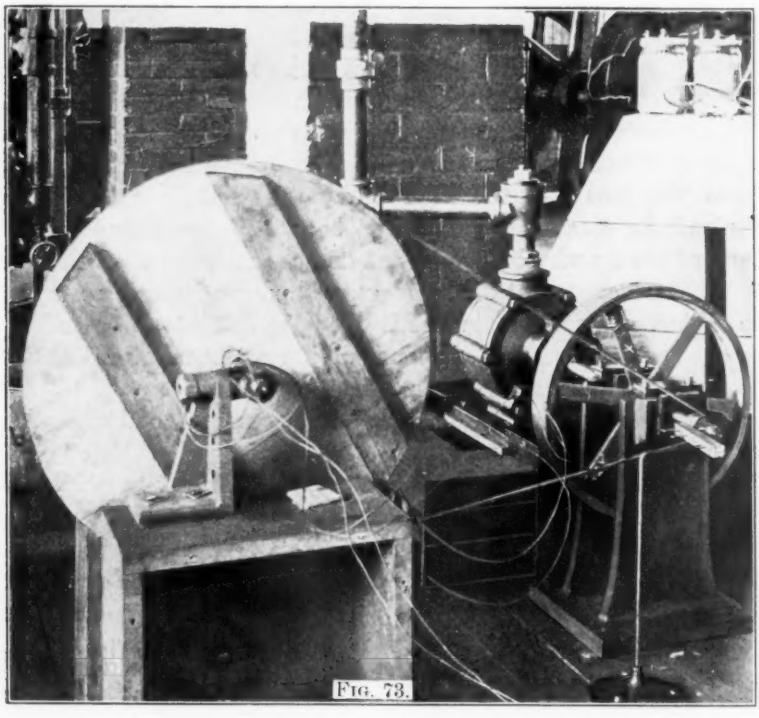


FIG. 73.

The wheels were all of cast iron and were clean, perfect castings. Two diameters were used, fifteen and twenty-four inches, and each wheel was a scale model of some actual fly-wheel designed by a reputable firm. The wheels numbered 1 to 10 had solid rims with the exception of No. 5. Wheel No. 11 was a special wheel, as will be explained later. The wheels numbered 12 to 17 had each two joints in the rim and were 24 inches in diameter. All the wheels numbered from 1 to 10 were reduced models of a solid-rim fly-wheel 10 feet in diameter now in use on

a 12 x 30 Allis-Corliss engine in the laboratory. The wheels numbered 12 to 15 were models of the same wheel on a larger scale, with rim joints designed by the writer. The two wheels numbered 16 and 17 were models of the fly-wheel of a Corliss blowing engine. Reference to the various cuts will show the general appearance of the various wheels tested before and after breaking. Tables I. and III. give the dimensions of the wheels in detail.

To give to the wheels the speed necessary for destruction, use

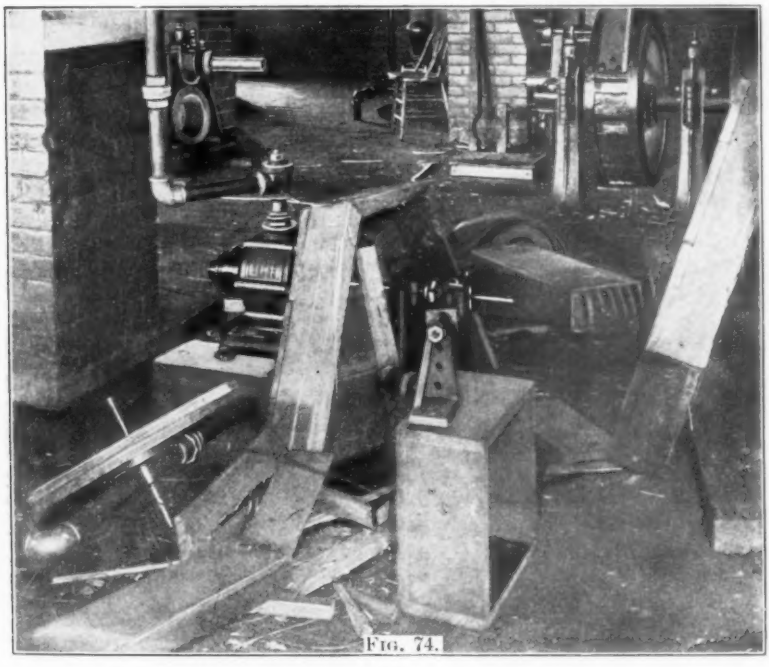


FIG. 74.

was made of a Dow steam turbine capable of being run at any speed up to 10,000 revolutions per minute. The turbine shaft was connected to the shaft carrying the fly-wheels by a brass sleeve coupling, loosely pinned to the shafts at each end in such a way as to form a universal joint, and so proportioned as to break or slip without injuring the turbine in case of sudden stoppage of the fly-wheel shaft. Fig. 73 shows clearly the arrangement of wheel and turbine.

One experiment with a shield made of two-inch plank (see Fig. 74) convinced us that safety did not lie in that direction, and

in succeeding experiments with the fifteen-inch wheels a bomb-proof constructed of 6 x 12 inch white oak was used. The first experiment with a twenty-four-inch wheel showed even this to be a flimsy contrivance, as may be seen by Figs. 75 and 76, from photographs taken immediately after the explosion of wheel No. 11. In all subsequent experiments a shield made of 12 x 12. inch oak was used, as shown in Fig. 77, which represents the

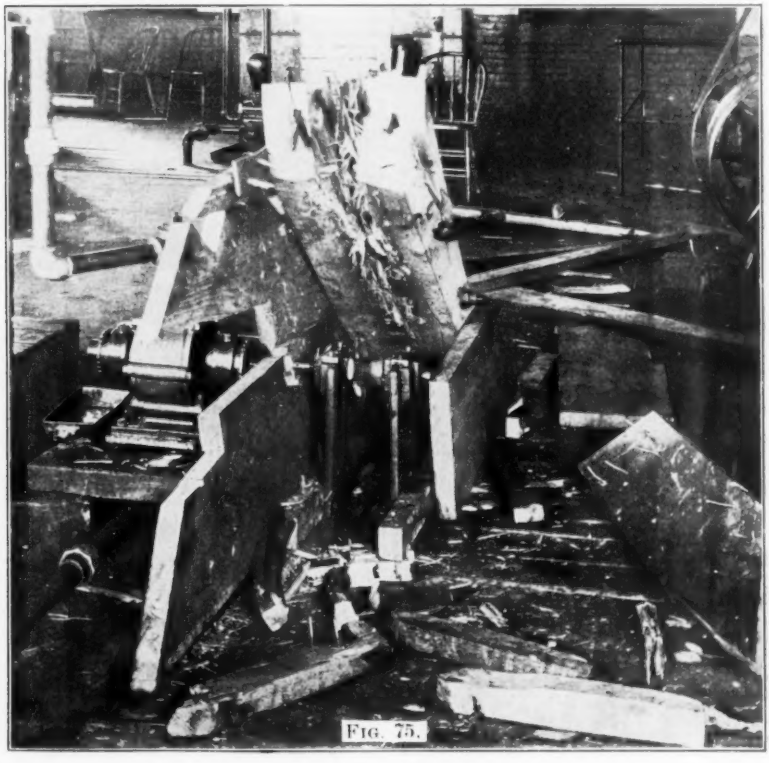


FIG. 75.

apparatus as employed on wheels Nos. 12 to 17. Even this shield was split repeatedly and had to be reënforced by bolts. The brick piers of the basement furnished havens of refuge for the experimenters, and no accidents occurred, but sundry holes in the brick wall, broken hangers, and riddled belting remain as souvenirs of the spiteful force of the flying fragments. The wheels were usually demolished by the explosion, as may be seen from the cuts. No crashing or rending noise was heard, only one quick, sharp report, like a musket shot.

The determination of the speed offered some difficulties at first, it being too great for the successful use of a counter or tachometer. A commutator of one break was arranged on the fly-wheel shaft, and this connected through the battery circuit with an ear-phone in an adjoining room. This arrangement worked satisfactorily, giving a clear, musical tone, and the number of vibrations corresponded closely to the speed as measured by a reducing counter shaft and speed counter. The commutator and the apparatus used in testing it are shown in Fig. 73. It was soon discovered that the audible tone produced by the machine itself

FIG. 76.

when running at a high speed corresponded exactly to the tone in the ear-phone, and consequently the ear-phone was discarded. Two observers, having trained musical ears and provided with tuning forks, had no difficulty in determining the pitch within half a tone, the quarter tones being estimated. The error due to this method did not exceed five per cent., and was probably less than if an attempt had been made to get the speed with a tachometer. The bursting speed of the wheels having rim joints was too low to produce a musical tone with any distinctness, and it became necessary to resort to the tachometer. It was not deemed safe for the observers to apply the instrument directly

to the fly-wheel shaft, and a counter shaft, reducing the speed from two to three times, was employed. Wooden pulleys were used, connected by a band consisting of several thicknesses of electrical tape. Careful observations convinced the writer that there was no appreciable slip.

The tachometer used was of the usual rotary pendulum type,

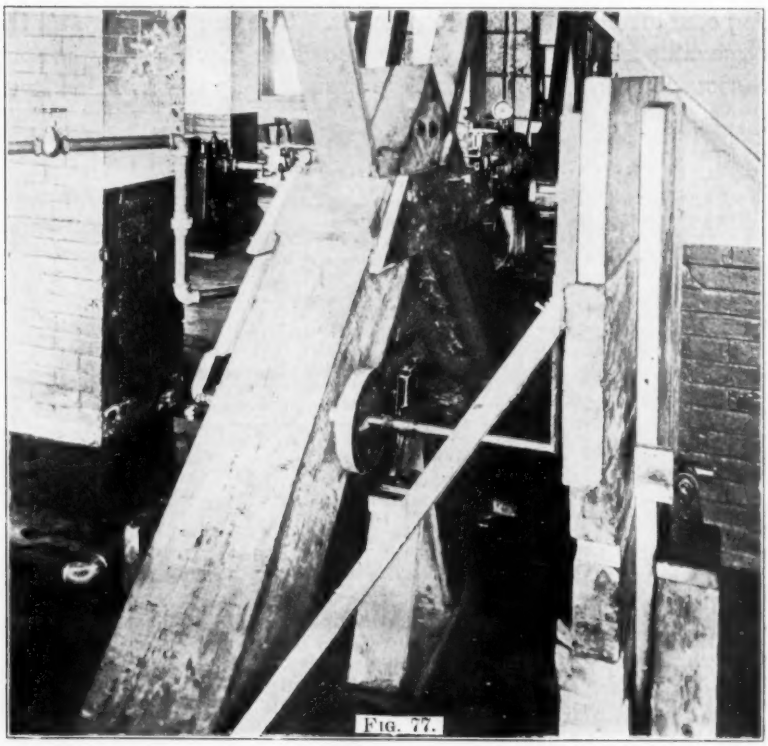


FIG. 77.

and was calibrated several times by comparison with a speed counter.

Fifteen-Inch Wheels.

Test pieces cast from the same ladle as these wheels were broken in the testing machine, and the following average values obtained for the breaking strength :

Tension, 19,000 pounds per square inch.
Cross-breaking, 39,000 pounds per square inch.

These wheels were all turned on the face and edges of the

rim, and were carefully balanced by winding copper wire around the arms near the rim.

Wheels Nos. 1 and 2 were practically identical in size and shape, as may be seen by Table I., and broke at the same speed, viz., 6,525 revolutions per minute, or a rim speed of 430 feet per second.

Fig. 78 shows the appearance of No. 2 after rupture.

Nos. 3 and 4 were similar to the preceding, save that the rims were turned down thinner. No. 3 burst at a speed of 6,035 revolutions per minute, or a rim speed of 395 feet per second, and No. 4 at 5,872 revolutions per minute, or 380 feet per second. The rims, being thinner, bent more between the arms, so that the

FIG. 78.

rims failed at a less speed. Fig. 79 shows the appearance of No. 3 after rupture. The shape of fracture at the outer ends of the arms in all the wheels usually indicated that the rim broke first midway between the arms, and that then the two parts of the rim flew outward and broke off at the arm.

Wheel No. 5 had two joints in the rim at opposite extremities of a diameter. The strength of the joint was designed to be one-third the tensile strength of the solid rim, but the wheel burst at only 2,925 revolutions per minute with a centrifugal tension of less than one-fourth that of the solid wheels.

Nos. 6, 7, and 8 had only three arms, every other arm having been removed from the pattern before casting. The object of this was to show more clearly the bending of rim due to

centrifugal force. These three wheels burst at speeds of 5,600, 6,200, and 5,709 revolutions per minute respectively. The figures for No. 6 were obtained with a tachometer and are doubtful, being probably too low. Wheel No. 6 is shown in Fig. 80.

FIG. 79.

Wheels Nos. 9 and 10 were of the original six-armed type, but with rims turned down to exceeding thinness, as shown in Table I. They each burst at a speed of 5,709 revolutions per

FIG. 80.

minute, or at rim speeds of 365 and 361 feet per second respectively, a reduction of over 16 per cent. from Nos. 1 and 2.

These results are summarized in Table II. An examination of the column containing the values of v in feet per second will show that as the segments of the rim between the arms become

weaker as beams, either through increase of length or decrease of thickness, there is a falling off in the bursting speed.

To determine to what extent the strength is affected by bending, values of $\frac{v^2}{10}$ have been calculated. As has been shown by Mr. Stanwood in the paper before referred to, this expression represents approximately the tensile stress on the square inch of section of rim due to the centrifugal force, for cast iron. By comparing these values with the tensile strength of the iron before noted, viz., 19,000 pounds per square inch, the amount of stress due to bending may be estimated. This difference

FIG. 81.

varies from 500 pounds per square inch in Nos. 1 and 2 to nearly 6,000 pounds per square inch in Nos. 9 and 10—being greatest in the wheels with thin rims or few arms. None of these wheels, however, except No. 5, would have been unsafe at the usual limit for fly-wheel rims of 100 feet per second. Wheel No. 10 would have had a factor of safety of over 12 at that speed.

Twenty-four-Inch Wheels.

All the wheels numbered from 11 to 17 were of the above diameter. No. 11 was a special wheel which had been in actual use. This wheel burst at 3,670 revolutions per minute, or a peripheral speed of 385 feet per second, which corresponds well with the average speed of the 15-inch wheels. The explosion was very violent, and completely wrecked the shield, as shown

in Figs. 75 and 76. The quality of the iron was unknown, save that it appeared clean and close-grained.

Flanged Joints.

The wheels numbered 12 to 15 were of the same model as the 15-inch wheels on a larger scale, but each wheel had two internal flange joints in the rim, midway between the arms, as shown

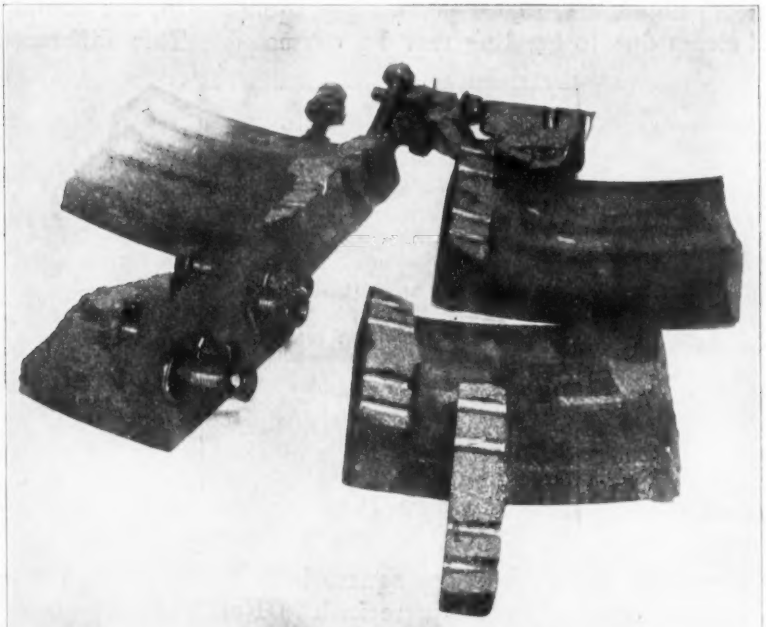


FIG. 82.

in Fig. 81. The joints were all carefully planed and the holes drilled to match. The wheels were not turned on the face, but were balanced the same as the others. In none of the experiments was there any shaking or tremor, only a dull roar of increasing intensity, then a single sharp report and quiet, except for the hum of the relieved turbine.

The proportions of the flanges and bolts are given in Table IV. The bolts used were of steel, and samples of each broken in the testing machine gave the results shown in the table.

Wheel No. 12 burst at a speed of less than 1,800 revolutions per minute, but the exact speed was not recorded. The flanges broke, as shown in Fig. 82, but the bolts were uninjured, except for a slight stretching.

No. 13 was a duplicate of No. 12 in every way, and burst at 1,760 revolutions per minute, or 184 feet per second. Fig. 83 shows the joints after rupture: the bolts uninjured, but the flanges broken through the bolt holes. The flanges of the pattern were then strengthened by adding $\frac{1}{4}$ inch to the thickness, the bolts remaining the same.

Wheel No. 14 burst at 1,875 revolutions per minute with a rim speed of 196 feet per second, in this case the bolts failing, as shown in Fig. 84. This cut is interesting as showing clearly the bending action of the centrifugal force on the bolts, using the inner edge of the flange as a fulcrum. This joint failed first; the wheel then opened like an oyster, fracturing the rim on each side of the joint opposite, the latter joint being left intact.

In No. 15, bolts $\frac{3}{8}$ inch in diameter were used, and the wheel failed at 1,810 revolutions per minute, or a rim speed of 190 feet per second. In this experiment the flanges of one joint were badly broken, as may be seen from Fig. 85, the bolts remaining whole. The second joint was uninjured, as in No. 14. The broken bolt seen in the cut came from the hub and should not appear. A close inspection of the two bolts in foreground of the cut shows them to be nearly sheared off at the joint, a phenomenon which is interesting, but which the writer is unable to explain. It would indicate that, as the bolts stretched and the joint loosened, one side of the joint was more affected by the centrifugal force than the other, causing a shearing action in the joint.

Fig. 86 shows the condition of the shaft and hub at the close of this experiment, a discouraging feature of this sort of work.

The results are summarized in Table V. It will be noticed that the rim speed is about one-half that of a solid wheel, and therefore the centrifugal tension about one-fourth. The joints in all the wheels were carefully made, and were relatively stronger than many joints in fly-wheels which are running to-day in our mills and shops. The centrifugal tension at the joint would be greater than that given in Table V. on account

FIG. 83.

FIG. 84.

FIG. 85.

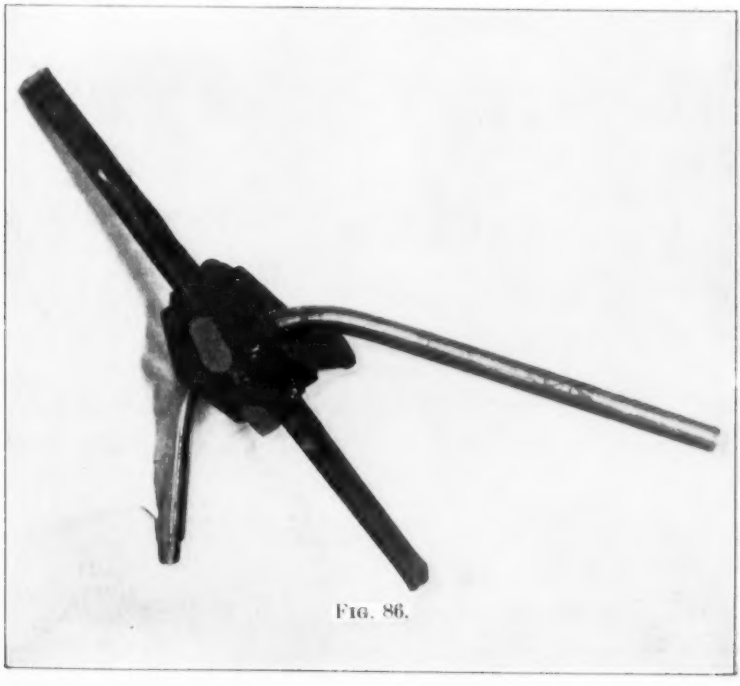


FIG. 86.

of the weight of flanges and bolts. At a rim speed of 100 feet per second these wheels would have a factor of safety of about $3\frac{6}{10}$, which is altogether too small.

Linked Joints.

Wheels numbered 16 and 17 were of the familiar rolling-mill type, with the joints connected by steel links over cast-iron lugs, as shown in Fig. 87, the links being heated and shrunk on. The dimensions of the lugs and links are given in Table VI.

Wheel No. 16 had three links to each joint, one on each face and one inside. This wheel burst at a speed of 3,060 revolutions per minute, or 320 feet per second. Each joint broke on one side, through the rim, without shearing the lugs or breaking the links, as may be seen from Fig. 88.

In No. 17 the link was omitted from the inner lug, leaving but two links to each joint. This wheel burst at a speed of 2,750 revolutions per minute, or 290 feet per second, in the manner shown in Fig. 89. On one side the rim broke as in No. 16 ; on the other side the lugs failed by breaking off. It is impossible to say which joint failed first. It is thus seen that No. 16, with three links, broke at a speed 65 per cent. in excess, and No. 17, with two links, at a speed over 59 per cent. in excess of that of the wheels with flanged joints.

From the summary in Table VII. it will be noticed that the strength of the rim at the weakest section is apparently in excess of the strength of the links, whereas it was the rim that failed in each case. It must, however, be remembered that the links were under direct tension, while the rim was subjected to bending in addition.

At 100 feet per second the factors of safety for Nos. 16 and 17 would be $10\frac{1}{4}$ and $8\frac{4}{10}$ respectively.

Conclusions.

1. Fly-wheels with solid rims, of the proportions usual among engine builders and having the usual number of arms, have a sufficient factor of safety at a rim speed of 100 feet per second if the iron is of good quality and there are no serious cooling strains.

In such wheels the bending due to centrifugal force is slight, and may safely be disregarded.

2. Rim joints midway between the arms are a serious defect and reduce the factor of safety very materially. Such joints

FIG. 87.

are as serious mistakes in design as would be a joint in the middle of a girder under a heavy load.

3. Joints made in the ordinary manner, with internal flanges and bolts, are probably the worst that could be devised for this purpose. Under the most favorable circumstances they have only about one-fourth the strength of the solid rim and are particularly weak against bending.

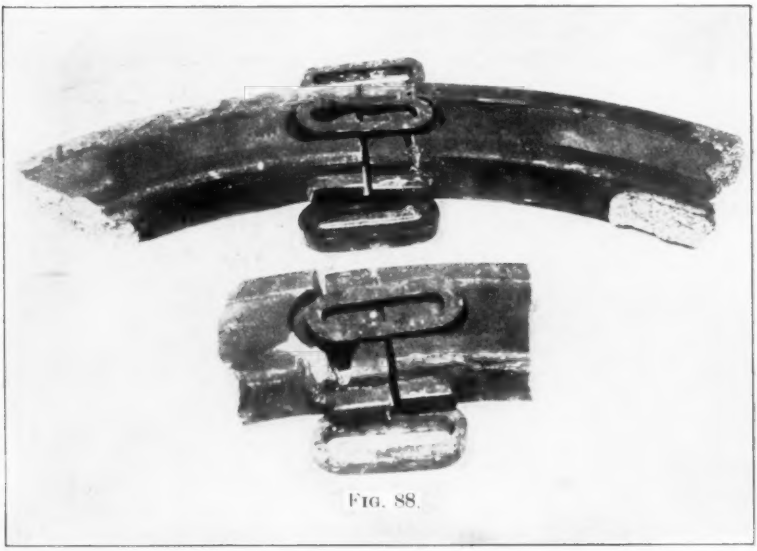


FIG. 88.

In several joints of this character, on large fly-wheels, calculation has shown a strength less than one-fifth that of the rim.

4. The type of joint exemplified in Nos. 16 and 17 is probably the best that could be devised for narrow-rimmed wheels not intended to carry belts, and possesses, when properly designed, a strength about two-thirds that of the solid rim.

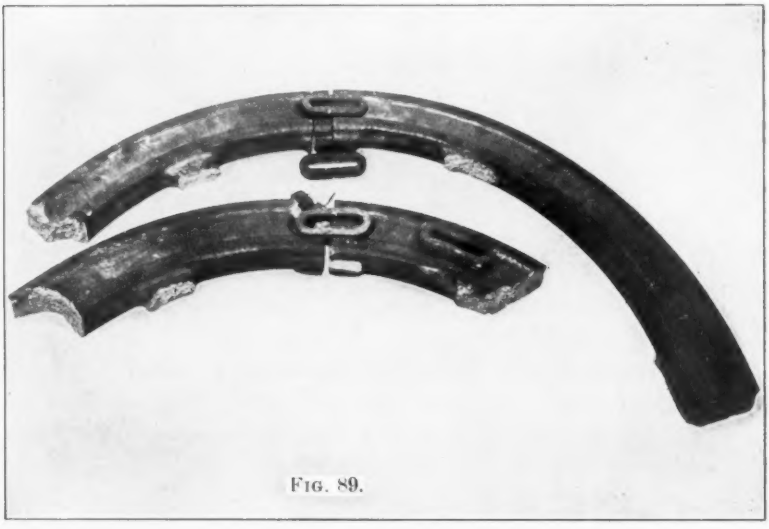


FIG. 89.

It is gratifying to notice the fact that since the subject of joints in fly-wheel rims has been so thoroughly ventilated during the discussions before this Society, several of our prominent engine builders have changed the designs of their wheels by bringing the rim joints opposite the ends of the arms.

The experiments which have just been described, although at times a trifle too exciting, were interesting from first to last. The writer hopes to supplement them by others on models of the more recent rim joints, and would be glad to receive any suggestions.

The more this subject is agitated, the less shall we have occasion to mourn the destruction of life and property on account of faults in the design of this most necessary element of the steam engine.

TABLE I.

FIFTEEN-INCH WHEELS.

No.	Rim.					Arms.		Weight of Wheel, Pounds.
	Style.	Diameter, Inches.	Br'dth, Inches.	Depth, Inches.	Area, Sq. Inches.	No.	Area, Sq. Ins.	
1	Solid	15 1/8	2	.70	1.4	6	.46	20.87
2	"	15 1/8	2	.65	1 3	6	.46	20.44
3	"	15	2	.615	1 23	6	.46	19.12
4	"	14 3/16	2	.52	1.04	6	.46	16.62
5	Sectional	15 3/32	..	....	....	6	.46	20.87
6	Solid	15 3/32	2	.69	1.38	3	.46	19.25
7	"	15	2	.615	1.23	3	.46	16.56
8	"	14 3/4	2	.475	.95	3	.46	13.68
9	"	14 3/8	1 7/8	.400	.75	6	.46	12.68
10	"	14 1/2	1 3/8	.347	.65	6	.46	13.00

TABLE II.

FIFTEEN-INCH WHEELS.

No.	Bursting Speed.		Centrifugal Tension $= \frac{v^2}{10}$.	Remarks.
	Revs. per Minute.	Feet per Second $= v$.		
1	6,525	430	18,500	
2	6,525	430	18,500	
3	6,035	395	15,600	Thin rim.
4	5,872	380	14,400	" "
5	2,925	192	3,760	Joint.
6	5,600*	368	13,600	Three arms.
7	6,198	406	16,500	" "
8	5,709	368	13,600	" "
9	5,709	365	13,300	Thin rim.
10	5,709	361	13,000	" "

* Doubtful.

TABLE III.

Twenty-four-Inch Wheels.

No.	Shape and Size of Rim.					Weight of Wheel, Pounds.
	Diameter, Inches.	Breadth, Inches.	Depth, Inches.	Area, Sq. Inches.	Style of Joint.	
11	24	$2\frac{1}{8}$	1.5	3.18	Solid rim.	75.25
12	24	$4\frac{1}{16}$	.75	3.85	Internal flanges, bolted.	93.
13	24	4	.75	3.85	" " "	91.75
14	24	4	.75	3.85	" " "	95.
15	24	$4\frac{1}{16}$	.75	3.85	" " "	94.75
16	24	1.2	2.1	2.45	Three lugs and links.	65.1
17	24	1.2	2.1	2.45	Two " " "	65.

TABLE IV.

Flanges and Bolts.

No.	Flanges.			Bolts.		
	Thickness, Inches.	Effective Breadth, Inches.	Effective Area, Inches.	No. to Each Joint.	Diameter, Inches.	Total Tensile Strength, Pounds.
12	$\frac{11}{16}$	2.8	1.92	4	$\frac{5}{16}$	16,000
13	$\frac{11}{16}$	2.75	1.89	4	$\frac{5}{16}$	16,000
14	$\frac{15}{16}$	2.75	2.58	4	$\frac{5}{16}$	16,000
15	$\frac{15}{16}$	2.5	2.34	4	$\frac{8}{8}$	20,000

By Testing Machine.

Tensile strength of cast iron = 19,600 pounds per square inch.
Transverse strength of cast iron = 46,600 pounds per square inch.
Tensile strength of $\frac{5}{16}$ bolts = 4,000 pounds.
Tensile strength of $\frac{8}{8}$ bolts = 5,000 pounds.

TABLE V.

Failure of Flanged Joints.

No.	Area of Rim, Sq. Ins.	Effect. Area Flanges, Sq. Ins.	Total Strength Bolts, Lbs.	Bursting Speed.		Cent. Tension.		Remarks.
				Rev. per Min.	Ft. per Sec. $= v.$	Per Sq. In. $= \frac{v^2}{10}$.	Total Lbs.	
11	3.18	....	.....	3,672	385	14,800	47,000	Solid rim.
12	3.85	1.92	16,000	....	...	....	.....	Flange broke.
13	3.85	1.89	16,000	1,760	184	3,400	13,100	" "
14	3.85	2.58	16,000	1,875	196	3,850	14,800	Bolts broke.
15	3.85	2.34	20,000	1,810	190	3,610	13,900	Flange broke.

TABLE VI.

LINKED JOINTS.

No.	Lugs.			Links.				Rim.	
	Breadth, Ins.	Length, Ins.	Area, Sq. Ins.	Number Used.	Effect. Breadth, Ins.	Thickness, Ins.	Effective Area, Sq. Ins.	Max. Area, Sq. Ins.	Net Area, Sq. Ins.
16	.45	1.0	.45	3	.57	.327	.186	2.45	1.98
17	.44	.98	.43	2	.54	.380	.205	2.45	1.98

BY TESTING MACHINE.

Tensile strength of cast iron = 19,600.
Transverse strength of cast iron = 40,400.
Av. tensile strength of each link = 10,180.

TABLE VII.

FAILURE OF LINKED JOINTS.

No.	Strength of Links, Lbs.	Strength of Rim, Lbs.	Bursting Speed.		Cent. Tension.		Remarks.
			Rev. per Min.	Ft. per Sec. $= v$.	Per Sq. In. $= \frac{v^2}{10}$.	Total.	
16	30,540	38,800	3,060	320	10,240	25,100	Rim broke.
17	20,360	38,800	2,750	290	8,410	20,600	Lugs and rim broke.

DISCUSSION.

Mr. Arthur J. Frith.—The paper by Mr. Benjamin is exceedingly interesting, emphasizing faults in fly-wheel construction, to which our attention has been previously called by Mr. Stanwood and Professor Lanza in our own Society, and by Professor Unwin in his treatise on the subject. The illustrations of fly-wheels and joints actually destroyed under known conditions are suggestive of ideas and remedies, and Mr. Benjamin's remark, in his concluding paragraphs, that he would be glad to receive any suggestions for further experiments is my excuse for this discussion.

Professor Unwin points out that in fly-wheels the centrifugal force tends to bow the rim between the arms and gives formulæ for determining the varying tensions in the rim, the extension of

the arm, and the bending moments of the rim caused by this inward pull.

The mathematical complexities of these formulæ, however, are not inviting, and all refer to the strains caused in a circular rim of uniform section by its efforts to assume a form for which it was not designed. Would it not be more rational, therefore, so to shape the curve of the rim that it will keep its shape under strain, and have the least possible tendency to cause bending moments? This is especially desirable in cast-iron construction, a material which is ill adapted to withstand bending. To be brief, if the rim were perfectly flexible, such as chain, and uniformly loaded with the centrifugal force, it would assume between the arms the shape of a modified catenary, probably an ellipse of which the versine of one-half the angle of the arms is the deflection, and of which the arms are the piers.

In a six-armed wheel the angle of the curve with the vertical at the pier line is 60 degrees. If, however, we assume any catenary for our curve with a known pier angle of, say, 55 degrees, then knowing the direction and amount of centrifugal force at the intersection of the arms and rim, we easily deduce, from a parallelogram of the forces, the strain in each arm due to the centrifugal force of the rim, to resist which, it and its fastenings should be figured in addition to that due to change of speed and its own centrifugal force. Between the arms the rim will assume the catenary curve only if uniformly loaded. This has a quicker curve at the centre, and is flatter at the ends. If loaded more heavily near the piers, the result would be quicker curvature at the pier and less towards the centre.

Is it not possible, therefore, in a solid wheel to lay out from the arms a parabolical curve with the pier angles slightly less than 60 degrees? Turn the outer curve of the rim circular and balance the area of the rim across the curve of strain. This would make it slightly wider at the arms, but the area could be kept the same by coring or cutting out at the web. Such a wheel would have no tendency to change its shape under strain, as with equal unit strains in the rim and arms they would stretch together. Professor Unwin, in the formula of the rim tensions, shows them increasing towards the arms; but I have seen no suggestions that the area of material should increase with the tension. No material error would arise by calculating the centre rim tension from a parabolical curve used in rope suspension, taking the chord as the span

and the versine as the deflection. The formula is exceedingly simple, and in a six-arm wheel the result tallies with the regular method.

In the case of a built-up wheel (Fig. 90), if the outer line of the rim be circular, while the inner line between the arms be parabolical or elliptical, with a deeper deflection than the versine, then the centre tension in the rim calculated by the parabolical formula would be greatly reduced, and by intelligent arrangement of the

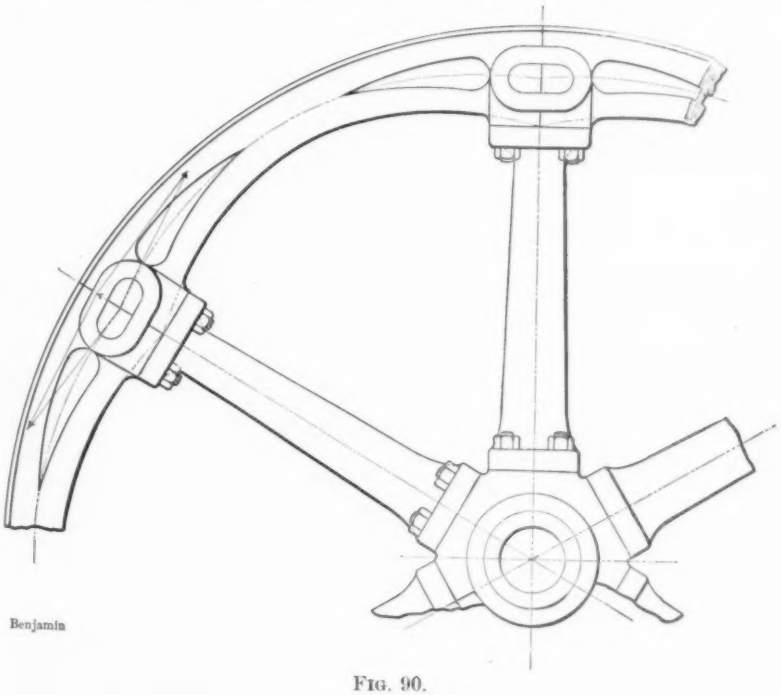


FIG. 90.

material the curve of resistance could be made to lie between the inner and outer rim and to pass directly through the centre of the fastening; thus it would have little tendency to change its shape. The joint would be placed over the arms, where the increased depth allows us to design a fastening which has the full strength of the centre of the rim, an advantage of very first importance; indeed, a riveted wrought-iron plate could be made to follow the inner curve of the rim to resist the entire tension, simply carrying the cast-iron centres as a load. In a later paper I intend to further develop this idea of determining the fly-wheel strain, with tabular

statements of tensions and such methods of determining the curves as I may be able to develop.

I should like to refer to a fly-wheel to which my attention was recently called. It was a built-up wheel with a broad face for a belt, and of large diameter. The centres were joined midway between the arms and the inner side of the rim where it overhung the arms, and between were stiffened with shallow ribs arranged in squares. The engine which drives this wheel broke its governor belt and raced, but was stopped before any damage occurred. Afterwards it was discovered that the wheel was running out of true, and an examination showed that out of sixty-two intersections of ribs and arms that could be broken by bending, thirty-one were cracked. As it does not seem reasonable to suppose that the wheel had started to break and had been stopped before the fracture could be completed, is it not likely that the rim, rendered more flexible by the fractures referred to, was able to assume the natural curve of the strains, and that after the accident the wheel was less liable to further fracture than before the ribs had parted?

In short, take such a section of wheel as that shown. The rim is perfectly circular and held in by arms through the centre. A circular rim will only keep that curve as long as all the strains are in the rim. As soon as this speeds up, the rim stretches, but the arms do not stretch; hence the rim, being held at points, bows out between. Strains are caused in this portion of the wheel from the attempt of the wheel to change its shape from that for which it was designed. In such a case as this the portion of the wheel between, with the four ribs, is held down by the arm; therefore these ribs would tend to crack in order to allow the rim to bend into the new shape. As practically 50 per cent. of all these ribs showed incipient cracks, probably just enough gave way to let the band section assume the curve that it ought to have had if the entire rim was to be placed in tension. In addition to this, you take the portion that is overhung—that, when it is speeded up, tends to stretch, and it has no arm to hold it back; therefore it stretches and goes further out from the axle than the central post. That would tend to bend these side ribs upwards, and we find that every one of these ribs has the incipient crack where, theoretically, we would expect it. Therefore the example is a good illustration of what I may be permitted to call the rational theory, but does not, to me, present anything remark-

able, being exactly what we ought to have expected. If we are going to build wheels of a broad face, and expect them to be held in towards the hub by arms, we must fasten the arms the full width of the rim. In other words, if, as I have been trying to bring out, we should figure the rims of fly-wheels not as you would figure a boiler, but as a curve that is under strain from the radiating lines of force, each section takes the position of a pendent rope. But the first condition of a suspension bridge is that it shall have piers to support it; a condition which, in a wheel of this kind, is absent. For instance, a suspension bridge might be made by taking a piece of boiler plate, bending it at the proper curve, and properly loading it. What would be thought if some one should then support that plate at each end by a square pier right in the middle of its width? It would be absurd, because it would throw the strain on the centre section of the plate. Now, this is what occurs almost universally in the construction of pulleys and band fly-wheels. We put a rim out here, which is to withstand the centrifugal force radiating from the centre when disturbed by the arms, and support it at one point midway of its breadth, instead of across the entire width of the rim.

Prof. Forrest R. Jones.—The lines of experiments which have been taken up by Professor Benjamin are especially valuable, since they are in a direction leading toward the improvement and increased safety of what has become, on account of the high speeds of recent years, a somewhat dangerous part of machinery. The strengthening effect of the arms of a fly-wheel seems to be shown clearly by the higher speed at which those having six arms broke in comparison with that of those having three arms. That the pulling-in action of the arms on the rim reduces the circumferential tension in the rim due to centrifugal force is unquestionably true; hence increasing the number of arms reduces the tensile stress in the rim, on account of reducing the tension due to centrifugal action, as well as reducing the bending stress due to the springing out of the rim between the arms. With a very great number of arms their action upon the rim would be almost the same as that of a solid web, with which the rim might be cut into numerous sections and still run at high speed without fracture.

Since the author of the paper invites suggestions it does not seem out of the way to call attention to the fact that experiments made under conditions similar to those which exist in operating fly or band wheels would be still more valuable than those already

performed. In the ones already made it appears that the speed of rotation was practically uniform, or at least slowly and uniformly increased, and that there were no sudden fluctuations of speed such as occur in all fly-wheels to some extent, and is especially marked in those applied to rolling-mill machinery and frequently in wheels which are attached to electric generators for power service, where the load may be suddenly increased by the throwing into operation of several motors almost simultaneously or by short circuits and grounds. It is a *prima facie* fact that greater stress will be thrown upon some parts of a fly-wheel when suddenly checked, as by the operation of a brake to its shaft while rotating at a given speed, than if it were allowed to rotate uniformly at that speed. And it might often occur that this additional stress would come in some part already highly stressed, thus causing rupture, which would not occur with the uniform rotation of the wheel. It would, therefore, seem that experiments conducted by bringing the wheel up to some predetermined speed, and then checking its rotation by the application of a brake to its shaft, would give results nearer to the actual safe speeds than those obtained by uniform rotation.

The conditions which are commonly met in the practical use of band wheels could be paralleled experimentally by having an auxiliary pulley with a belt running over it and the band wheel under test, and varying the resistance to rotation of the driven pulley by a brake applied to its shaft. It seems probable that the wheel under test would often be broken in this way at a much slower speed than when rotating uniformly.

Doubtless Professor Benjamin has thought over all this matter, and has realized the increased danger and expense of conducting experiments more complicated than those already made. And probably from his experience he has realized more fully than any one else that a rapidly rotating fly-wheel is about as dangerous as a charge of dynamite, with all the conditions favorable for its explosion at any instant, and that one may well hesitate about so complicating the apparatus as to require the presence of any one near it while in motion. But still this does not change the fact that experiments made under conditions practically analogous as those existing when rupture has accidentally occurred in wheels in actual use, would be the most valuable that could be made in the line of bursting tests upon fly and band wheels.

Mr. Wm. Wallace Christie.—Two or three years ago I con-

demned a fly-wheel about 4 feet 6 inches in diameter, for 12-inch double belt, turning 300 revolutions per minute, because it turned out of truth one-quarter of an inch at the circumference in a direction parallel to the shaft—or, in other words, "wobbled" one-quarter of an inch.

I would like to ask how much of a sidewise movement may be considered safe for given diameter, speed, and weight of wheel.

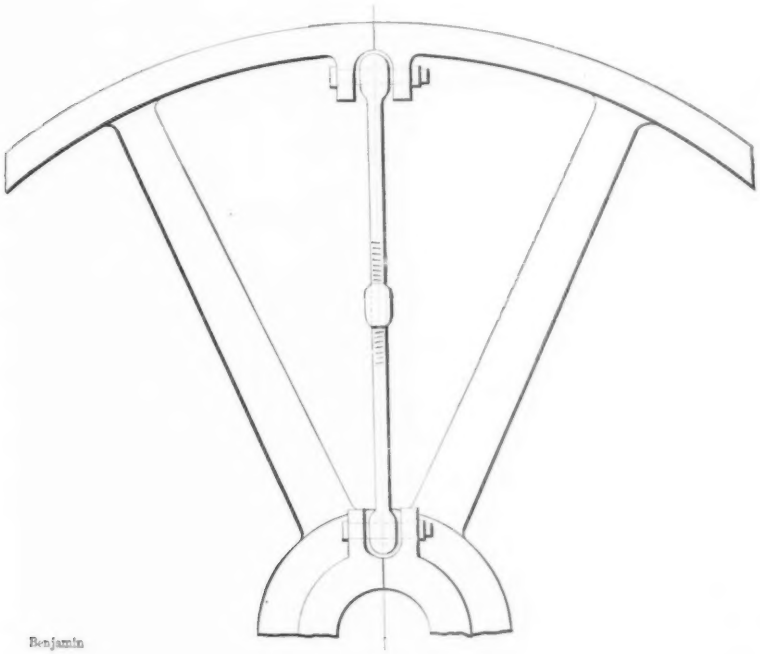


FIG. 91.

The wheel in question was taken out and turned true on its shaft, and is running at the present time.

Mr. C. Seymour Dutton.—Professor Benjamin's account of his experiments with cast-iron wheels is not only of general interest, but is, unquestionably, of real value to the designer of large wheels; although, as the author indicates, some allowance must be made for the fact that small castings are somewhat stronger and more reliable than larger masses of the same material.

It is seldom that the commercial consideration of first cost can be entirely neglected in the design or construction of any machinery, and the expense involved in fly-wheel rims of wire or steel

plates has prevented their general consideration. And yet, as Professor Benjamin observes, the mortality of cast-iron fly-wheels has become alarmingly common.

The failures being practically confined to wheels having jointed rims, seem to indicate that these joints are usually a greater source of weakness than is generally realized, and that solid cast iron rims of good design are sufficiently strong for any ordinary requirements.

Some of our large engineering works can make solid wheels of almost any desired size or weight; but, unfortunately, most cases are limited by the requirements of railroad transportation.

In consequence a common practice is to make solid wheels up to ten feet in diameter, from ten to twenty feet in halves, and exceeding twenty feet in segments.

Joints being thus unavoidable, they should be made as harmless

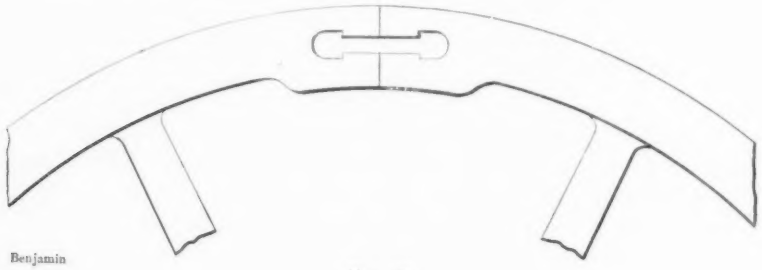


FIG. 92.

as possible. The tangential stress is, usually, easily provided for; but the radial force, if aggravated by the weight of the lugs or flanges for the connection of the rim midway between the arms, seems to cause nearly all the trouble, and is not so easily handled. Radial tension rods, from the hub to the joint (Fig. 91), are sometimes used to counteract this force, and if Professor Benjamin should continue his valuable investigation, I would suggest this device as a subject for a test.

Another form of rim joint about which I have always had some curiosity is that indicated in Fig. 92. This is intended for simple fly-wheels in place of the usual linked joint. The steel tie-piece extends entirely through the rim, which is reënforced on the inner side to preserve the section.

Possibly the best solution of the large-wheel problem, in many cases, may lie in the location of rim joints at the extremities of

the arms, but the construction of two joints in different planes at one point, and of the necessary strength, will not always be found wholly devoid of difficulty.

If any of the students of the Case School of Applied Science should feel disappointment about their absence from certain recent demonstrations in Manila Bay and near Santiago de Cuba, I would suggest that they might gain a considerable degree of consolation by assisting Professor Benjamin in his investigation of wheels.

Mr. J. B. Stanwood.—It occurs to me to ask the author that if he has further experiments of this nature to be made, he should determine the relative strength of wheels having their joints

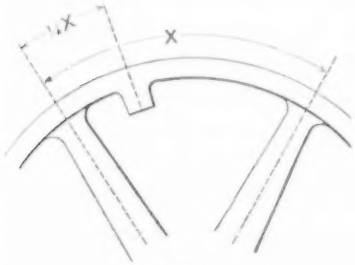
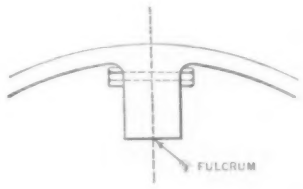


FIG. 93.



Benjamin

FIG. 94.

spaced at one-quarter the distance between the arms as compared with those having joints midway between (Fig. 93). Recently, in wheels made in halves, I have placed the flanged joint in this position, which corresponds closely to the plane of reversal of the flexure in a beam fixed at both ends and uniformly loaded. At such a plane there is no bending moment, but shear only.

In connection with the bursting of his wheel No. 14, Professor Benjamin shows how the inner edge of the flange acts as a fulcrum for the bending moment resulting from centrifugal force in that plane, and that in this case, with heavy lugs, the bolts failed first. In this connection I would add that I have been making the flanged lugs very deep and strong (see Fig. 94), in order to reduce the tension on bolts to a minimum. (The bolts should be as close to the rim as possible.) The thickness of the flange should be enough to resist bending moment.

It would seem that Professor Benjamin might undertake a series of investigations to determine the proper proportions for bolts and flanges of rim joints, also the proper location of such a

joint to give the maximum strength to the rim. I believe that an experimental determination of a joint of maximum strength to be very desirable.

Mr. McBride.—I want to say that I am very much interested in the wheel business now. I have some interesting data here on a wheel which I have in my mind's eye that I want to lay before the Society, and I hope they will be the means of bringing out some important points. I have no opinion whatever to offer on it myself. I simply submit the facts to the experts. I have three views of the wheel I speak of. This large one is a cross section through the rim at the point of the arms (Fig. 95). Fig. 96 is an inside view of the wheel looking from the centre out towards the circumference, showing the junction of the arms and the rim, and also the connection of the segments between the arms. Fig. 97 is a side view of one arm where it is attached to a segment. You will notice that the segments are connected

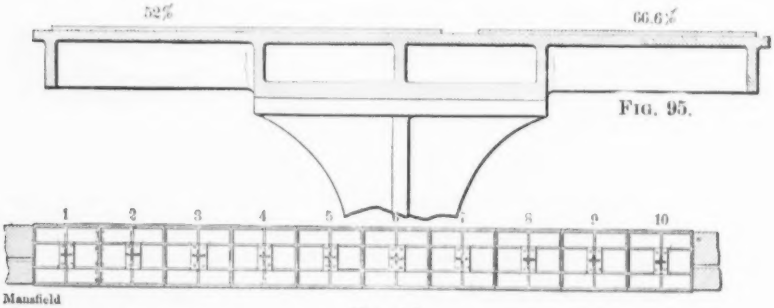


FIG. 95.

FIG. 96.

FIG. 97.

together with flanges and bolts. This wheel is of the following dimensions: 26 feet in diameter, 90½ inches on the face, and runs about 53 revolutions per minute. It has on it one 48-inch belt and one 34-inch belt. You will notice that this wheel has a very large overhang on each side outside of any support it can possibly get from the arms except through those brackets. The overhang of the 48-inch belt is 52 per cent. outside of the support it gets

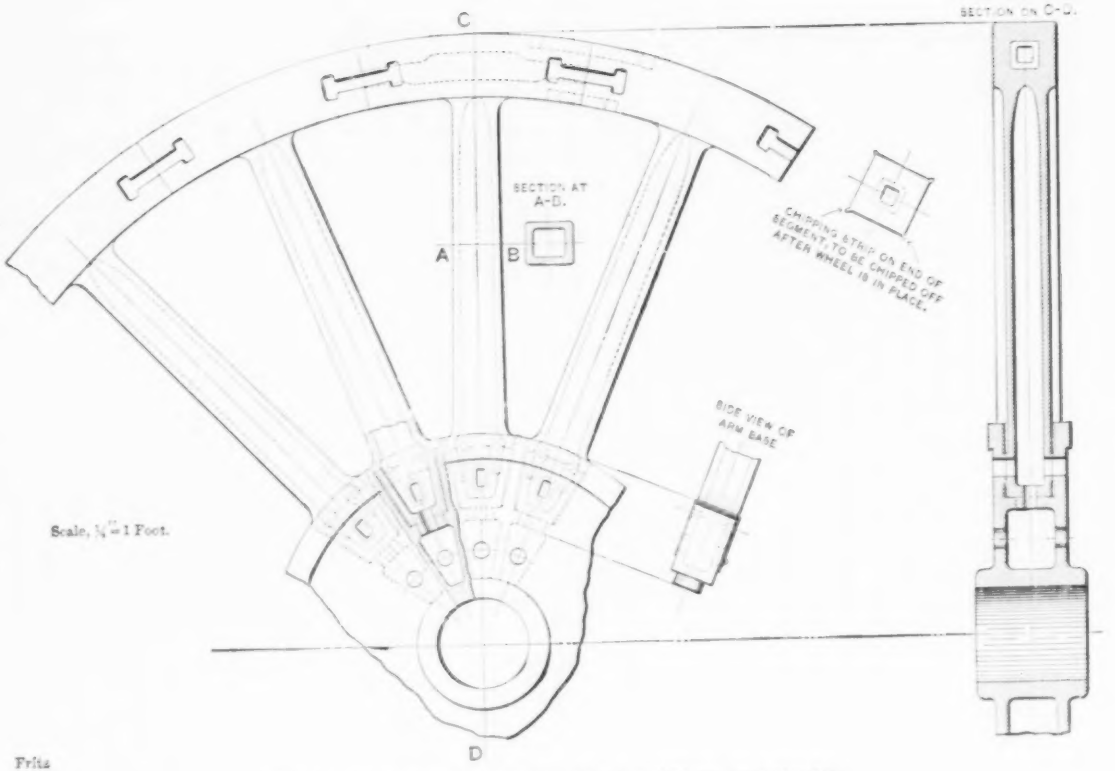
in the arms, and of the 34-inch belt, 66⅔ per cent., or sixty possible places to break where the arm is attached to the rim; that is, at this point around the flange where the flange of the end of the arm attaches to the flange of the rim, there are sixty places where this rib construction can crack or be broken. Thirty-three of those are broken, or 55 per cent. Of two hundred other places of similar construction which might be broken through shrinkage cracks, not a single one is broken—not one. Of the total breaks, twenty-one, or 63 per cent., are on the side of the wheel having the narrow belt. This is a 2-inch rib running out, and smaller ribs running around. Of the twenty-two ribs crossing the rim from the arms—that is, these points here—eight are broken, and seven, or 87½ per cent., are on the side having the narrow belt. Of the twenty-five cracks in the ribs running down, fourteen, or 56 per cent., are cracks on the side having the narrow belt.

Mr. John Fritz.—All my friends know that I am not a speaker, but as some reference has been made to rolling-mill fly-wheels going to pieces, I want to say that while it is a reflection upon the mechanical and metallurgical engineers who built them, I would rather build fly-wheels that would not go to pieces at any reasonable speed than to talk about them in public.

My method of designing fly-wheels for machinery is shown in the accompanying cut (Fig. 98), which represents a type of wheel for a 25-inch rail train running from 70 to 80 revolutions per minute. The wheel is 26 feet in diameter, and was first started in 1873, and was in constant use day and night until 1892 or 1893.

The subject of fly-wheels is of so great importance that I am going to ask to be allowed to present the methods used in getting up this wheel and in its fitting and finishing, together with some other drawings, as a paper for a future meeting of the Society.

Mr. McBride.—I see quite a number of fly-wheel makers here to-day and I would like to ask them, if it is not giving away trade secrets, if they pursue any method of using a certain factor of safety for fly-wheels; whether they design a wheel to run at 60 revolutions a minute, if you please; whether they make that strong enough to stand twice 60 or three times 60 or any other number of 60, or whether they simply make it strong enough to stand at 60 and take their chances on going to pieces. When a man builds a boiler or builds a bridge he tries to incorporate in it a certain factor of safety, and he makes it many times stronger than any load that is likely to be brought on it, and I would like



28" RAIL MILL & 33" BLOOMING MILL ENGINE, FLY-WHEEL.

FIG. 98.

to know if anything of that kind is pursued in the construction of fly-wheels.

Mr. Fritz.—I would like to say this with regard to the wheels: A very prominent engineer saw one of those wheels, and asked me: "Haven't you got that wheel stronger than there is any use for it?" I said, "I don't know; if it is, I will never know it."

Mr. Henning.—I have stood alongside of Mr. Fritz's wheels and know that they run pretty nearly true. I don't think I ever saw one run a sixteenth of an inch out of true, trying it by holding a point close to the pulley to see how it did run. Occasionally the engines did run faster than they ought to, but the engines were always stopped before the wheels gave way. I know the design Mr. Fritz has adopted has never given any trouble, because he recognized several things in construction which many engineers using cast iron, even at the present day, do not recognize, and from that point of view I should like to answer Mr. McBride's inquiry. The factor of safety, of course, is a certain constant, provided the quality of the material is known. There are no means, at the present time, of knowing what the actual strength a block of cast iron cast twelve inches square may be. Twelve inches square is a reasonable section for a fly-wheel rim. The outside metal has a certain strength, while the metal near the centre is very different. It may not have any resistance whatever, on account of piping or crystallization or internal strain. If the factor of safety were proper, there would be no more danger of the breaking of the fly-wheel, because there would be ample strength in every case. On the other hand, the elastic deflections in fly-wheels I believe are never calculated, and we know that the elastic changes of shape of fly-wheels are three times as great as those of steel and two and a half times as great as those of wrought iron. We know the modulus of elasticity of cast iron runs somewhere from nine millions to eleven and a half millions, while steel runs from twenty-seven millions up to thirty-one millions. Wrought iron is twenty-four millions to twenty-six millions. Now, the stretch of these arms is all dependent on that difference in figures from thirty-one millions down to nine millions, and the change of a shape of fly-wheel is exactly proportionate to that difference in moduli. Everybody considers that cast iron is the most rigid material in use. It is just the reverse. It is the most flexible and most elastic. That is, for the same load cast iron will stretch or change its shape three times as

much as steel and two and a half times as much as wrought iron. Therefore extra provision ought to be made, which is rarely made. What Mr. Fritz does by casting the rims hollow is this—he obtains thin sections, the strength of which is fairly well known.

Nothing is known of the actual strength of cast-iron beams having sections such as are used in solid rims of large fly-wheels. The strength of casting more than three inches thick is very uncertain, and vastly less than given by calculation of strength based on book constants. If the distortion of a wheel is prevented, just exactly as Mr. Fritz does, by vertical flanges, then it is stiffened in proportion to the cube of depth. Therefore it can thus be stiffened in a larger proportion than by adding weight in the shape of a solid block, making the rims strong enough. They must be sufficiently rigid to withstand the distortion in the other parts of the wheels. That, under abnormal conditions, is very great. Distortion is also due, not to the resistance of the cast iron, but entirely and alone to its flexibility or elastic distortion under loads, which is nearly three times as great as steel and two and a half times that of wrought iron. Therefore, in Mr. Fritz's wheel, which has very deep arms, he does the very thing that is necessary to make a wheel stiff and rigid. He designs for rigidity under loads. In wrought iron or steel it would be quite different. In using cast iron it should be used in shapes in which its rigidity will resist for elastic distortion under load and there will be no trouble. The trouble with nearly all fly-wheels that break is this, that the essential factors of strength, actual rigidity and resistance to change of shape, are not properly provided for, while they are provided for in the designs for which Mr. Fritz is responsible.

Mr. Jesse M. Smith.—There are two classes of fly-wheels: first, those that are used simply to carry the shaft around, and having a section of rim which is very thick and rather narrow. The second are the band wheels, having a very thin and wide rim. It is the band wheels which give the trouble. This section which Mr. Fritz has proposed is admirable for a mere inertia wheel, and enables him to get good material in the best shape to resist the strain which comes upon the wheel. But this wheel which is shown on the board is a band wheel and must, necessarily, be very wide. The shape of what are intended to be strengthening ribs can be modified so as to give a rim of uniform strength; that is, the ribs may be tapering from the arms, which is the point where the moment

of forces is greatest, to a point midway between the arms where the moment is least. Now, it has occurred to me that in the case where Mr. Benjamin places the joint midway between the arms (Fig. 81), and he increases the thickness of the flanges to get additional strength, he also increases the weights, so that this joint tends towards its own destruction, because the greater weight has the greater centrifugal force. It would seem, therefore, that if it is possible it would be better to have the joints at the arms where the centrifugal force can be directly counteracted by the tension of the arms.

I agree with Mr. Fritz that the hollow rim adapts itself admirably to the strains brought upon it by centrifugal force, and there is no reason, even in a band wheel, why the thickness of the inner and outer walls of the rim may not be varied at each section so as to exactly meet the strain at that section. I think, however, it would be a more expensive wheel to build than that with thin rim and ribs which we usually find in practice; but the hollow rim construction is certainly capable of producing a much stronger and better wheel.

Mr. Henning.—I would like to call attention to one statement that Mr. Smith made. Many wheels are designed for uniform resistance of every part. That is the very thing which is wrong. It is not uniform resistance—it is uniform rigidity which must be provided. If there is uniform resistance, then the parts which deflect more will, of course, affect the others, and it is not a question of resistance at all; it is a question of rigidity. If the wheels were designed for uniform rigidity under the varying conditions of the material in them, then there would be less liability to fracture.

*Prof. Chas. H. Benjamin.**—I am glad that this paper has brought out such a full discussion and so many valuable suggestions as to further experiments.

The suggestion of Professor Jones in regard to sudden changes of speed is very worthy of consideration. It is difficult, however, to effect such changes with safety to the operator.

Whatever belts, secondary pulleys, etc., are used would need to be included in the shield and would all be destroyed at each experiment.

It may, however, be possible to stop the shaft of the fly-wheel itself rather suddenly by means of a brake acting on this shaft outside the shield.

* Author's closure, under the Rules.

It seems to me that the joint suggested by Mr. Stanwood and shown in Fig. 93, *i. e.*, the joint at the point of inflection of the rim, is one solution of the difficulty.

I am having some model wheels constructed with joints so located, and intend to test them the coming year. It is to be noted that moving the flange also changes the point of inflection. The enlargement of the flange shown in Fig. 94 of Mr. Stanwood's discussion I consider a mistake, as the additional weight thus concentrated at the joint will more than neutralize any advantage gained. It must also be remembered that the flange is likely to be broken at the bolt holes by the bending moment due to the pressure at the fulcrum (Fig. 94).

I am having made several models of wheels with various forms of link and "prisoner" joints, such as are used in rolling mill and blowing engines. There is no reason why wheels of this character should not be amply strong. The section of the rim is that best calculated to resist bending, and admits of reliable connections at the joints in the direct line of stress. The two experiments on link joints in the paper show the comparative safety of this type. It is the fly band wheel with its wide, thin rim which presents the more difficult problem, and it is on this type of rim that there is need of further experiment.

In this connection it might be well to mention the danger from generator pulleys which run at a high speed and are balanced by bolting heavy weights on the inside of the rim at its thinnest section.

The rim is thus further weakened by the hole which is drilled for the rivet or bolt, and the balance weight adds to the centrifugal force without adding to the strength. An example of this is shown in the December number of *Power*.

I am sorry that none of the members have offered an explanation of the shearing force which is so apparent in some of the flange joints, nearly shearing the bolts in two in one instance (see p. 219, and Fig. 85). It is generally supposed that midway between the arms is a point of maximum bending moment and zero shear.

I hope to make another series of experiments during the coming year on various shapes of rim and styles of joint, including wheels with steel spokes.

If it proves feasible to photograph some of these wheels at the instant of rupture, the pictures thus obtained will probably throw still more light on the problem.

DCCCI.*

THE MECHANICAL PLANT OF A MODERN COMMERCIAL BUILDING.

BY WILLIAM H. BRYAN, ST. LOUIS, MO.

(Member of the Society.)

THE modern commercial building is, in many respects, an engineering structure. The problems of foundations, design of frame and walls for the desired loads and wind pressures, the plumbing and sanitary arrangements, call for engineering skill of the highest order. This paper, however, will be confined to the discussion of the mechanical, steam, and electrical plants, which are of fully equal importance.

It is now generally admitted that no features of the modern building stand in greater need of consideration by specialists. Such treatment has always been desirable, and is more so to-day than ever, on account of the wide variety of service required and the improvements which are constantly being made.

These considerations involve first of all:

The *Boiler Plant*, which necessitates the determination of the average and maximum requirements of all the steam cylinders and other steam-using apparatus in the building. It also involves consideration of the characteristics of the fuel and water to be used, the selection of the type of boiler, the smoke flue and chimney, and usually the prevention of smoke.

Next in importance comes the question of *Power* development and distribution; its average and maximum amounts; whether steady or fluctuating. Which of the many types of modern steam engines shall be adopted; the number and size of units; whether high or low speed; simple or compound; condensing or non condensing; direct-connected or belted? Shall the power be distributed by shafting, belting, rope-drive, electricity, compressed air, or what not?

* Presented at the New York meeting (December, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

The *Lighting* system comes next in importance. In these days illumination by electricity is almost a foregone conclusion. Shall the current be supplied from a central station, or generated within the building? Shall the lighting be by arcs or incandescents; or if in combination, how distributed? Shall motors be driven from the same apparatus? What system of distribution shall be employed; what voltage—110 or 220; what character of current—direct or alternating? If an isolated plant is installed, what shall be the number and size of units; shall they be direct-connected or belted; what as to their general characteristics? Shall the units be independent, or shall they be connected by some scheme of shafting or clutches? What switchboard facilities are necessary? What indicating and recording instruments, switches, safeties, etc.? Shall the arc lamps be of the open or enclosed type, operated in series or multiple? What voltages, efficiencies, and candle powers shall be employed for the incandescents? What shall be the general character of the wiring, its percentage of drop, and its subdivision into sections for independent control by switches?

The *Elevator* system is a feature of prime importance. How many elevators shall there be; where located? How many passenger, and how many freight; how many auxiliaries, such as dumb waiters, ash hoists, etc.? What shall be the elevator loads and speeds? How shall they be driven? If by steam, whether belted or direct? If of the hydraulic pattern, shall they be high or low pressure; with horizontal or vertical cylinders? What type of pumping machinery shall be used—the old style direct-acting duplex with simple high-pressure cylinders; the same engine fitted with compound cylinders; the simple crank and fly-wheel pattern; or the more modern compound or triple-expansion crank and fly-wheel "high duty" engine? Shall it be provided with condensers? If the elevators are to be operated by electricity, at what voltage? Direct or belted; how counterbalanced, and provided with what automatic and safety devices?

The *Heating and Ventilating* systems are by no means of least importance. Shall the system be hot air, hot water, or steam, or a combination of them? If hot air or indirect, shall it be by the natural, or hot blast system? If hot water, shall it be natural circulation, or by pumping? If by steam, what shall be the pressure carried, the type of radiation, its amount and dis-

tribution? What system of piping shall be employed? How many changes of air per hour are required, and shall it be by natural or forced circulation?

Very closely allied to the heating plant is the *Steam Pipe* system connecting the boilers, engines, pumps, heaters, receivers, etc., with their accompanying valves, regulators, covering, blow-offs, etc., forming in a large building a very complicated system, permitting the utilization of the exhaust from the engines and pumps for heating. The general arrangement of this work, and the selection of the proper apparatus, all demand the most careful study.

The above may serve to indicate that the problems under consideration involve a wide range of engineering experience and research, and necessitate the reaching of conclusions in regard to many matters on which there is room for divergence of opinion.

Each type of building must, of course, be considered as an independent problem. The mechanical requirements of the modern office building, for instance, are quite different from those of the modern hotel or apartment house. Both differ radically in their requirements from business houses. Even here the retail store involves considerations quite different from the wholesale store or warehouse. Peculiarities of trade affect even different houses in the same general line. It is impossible, therefore, to draught general rules. Each installation must be studied with reference to its own peculiarities.

The author submits with some diffidence the methods employed in designing the equipment for a large building recently constructed in St. Louis. It is not claimed that they involve anything particularly original, or that any special or unusual difficulties were encountered. It is offered simply as an example of what is believed to be good current practice. In some respects the solutions reached may appear novel, and the combination of so large a number of desirable features in a single plant may be said to have marked an era in this field of work in this section of the country. It is believed that in adaptability to somewhat peculiar conditions of service, and in low first cost, as well as low cost of operation and maintenance, the installation is at least interesting.

A front view of the building is shown in Fig. 99. It is a brick and stone building, of the character known as "standard slow

combustion," as defined by the Board of Fire Underwriters. It is owned by the Commerce Realty Company, and is situated on the north side of Washington Avenue, about midway of the block between Ninth and Tenth streets. Its ground space is 109 feet front by 225½ feet deep, and it is eight stories high, besides basement. Fig. 100 is a basement plan; Fig. 101, first floor; Fig. 102, a characteristic upper floor; Fig. 103, the eighth or top floor, and Fig. 104 a cross section showing elevations and heights of stories. The tenants are the Hagadine–McKittrick Dry Goods Company, an old established wholesale firm, located for many years at the southeast corner of Eighth Street and Washington Avenue, occupying a building of less than half the floor area of the new one. These facilities having become inadequate for their business, arrangements were made in the fall of 1897 for the erection of a new building. The architects of the new building were Messrs. Eames & Young, of St. Louis, who, at an early stage of the work, called into association with them as advisory and consulting engineers the firm of Bryan & Humphrey. Ground was broken November 16, 1897, and the work was pushed urgently when the weather permitted, and for part of the time night and day. The tenants moved in on June 1, 1898, partial service from the mechanical plant being then available, and complete service being furnished shortly thereafter.

In the designing of this plant, a prime consideration—and one which always presents itself—was the necessity of getting a high-grade installation for as little first cost as possible. Owners, as a rule, object to paying an advance for machinery of higher efficiency, unless the results have been demonstrated so often as to be beyond question, and even then they hesitate, or expect the tenants to pay the difference. A distinction must, of course, be made between apparatus of established merit and of proven adaptability to the case in hand, and what may be termed "refinements"—good in themselves, but of doubtful value under the expected conditions of operation.

To begin with, the location and character of the building, and the work to be carried on in it, were discussed by the architects and engineers, in consultation with the owners of the building and their prospective tenants, including the heads of the different departments and floors, and their operating engineer. Having determined on the general character of the plant as

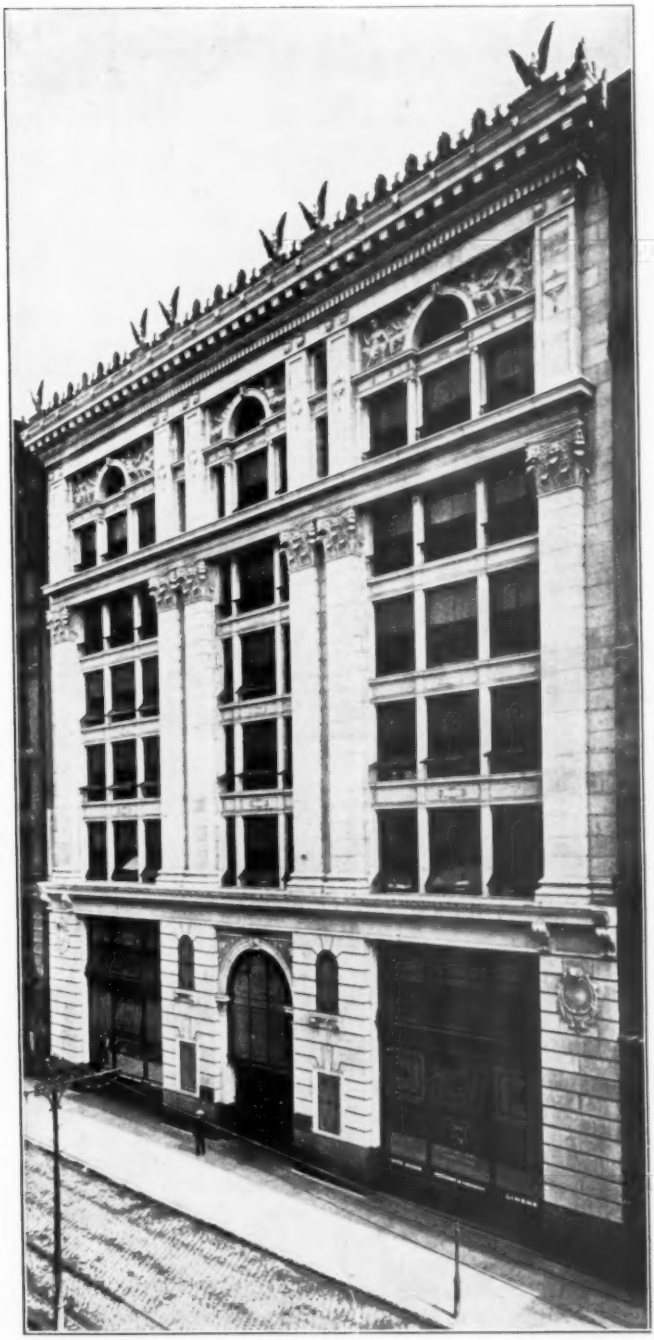


FIG. 99.

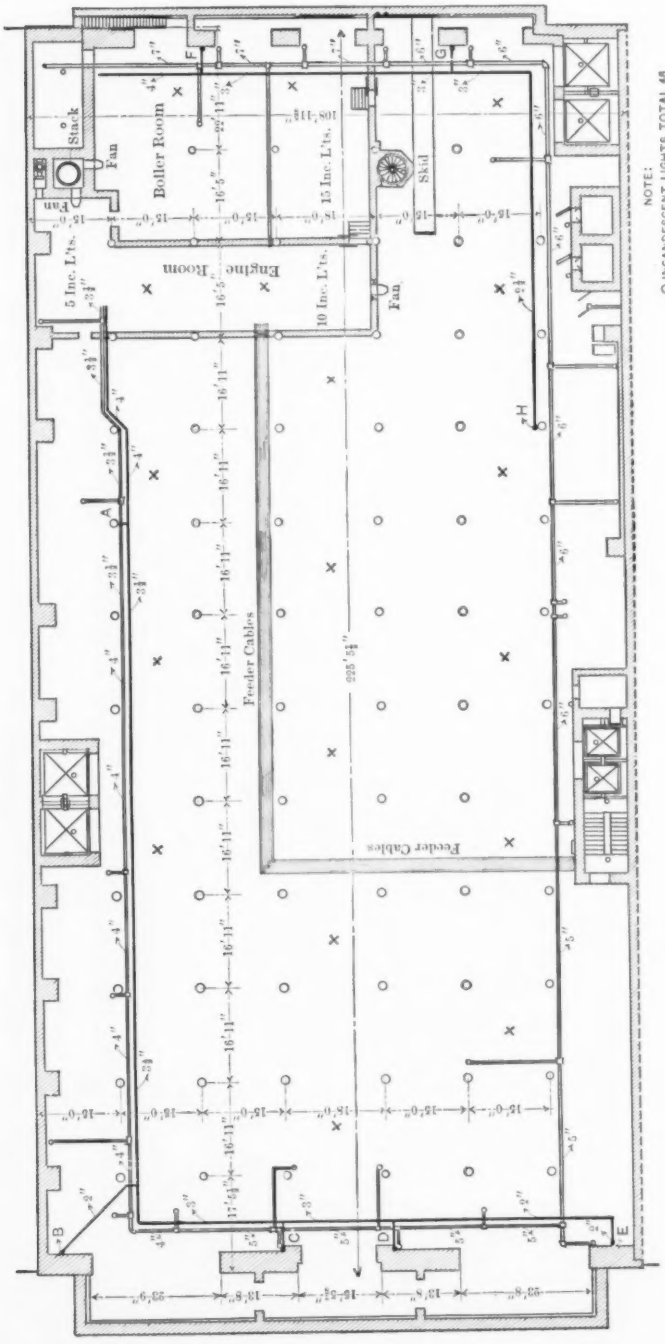
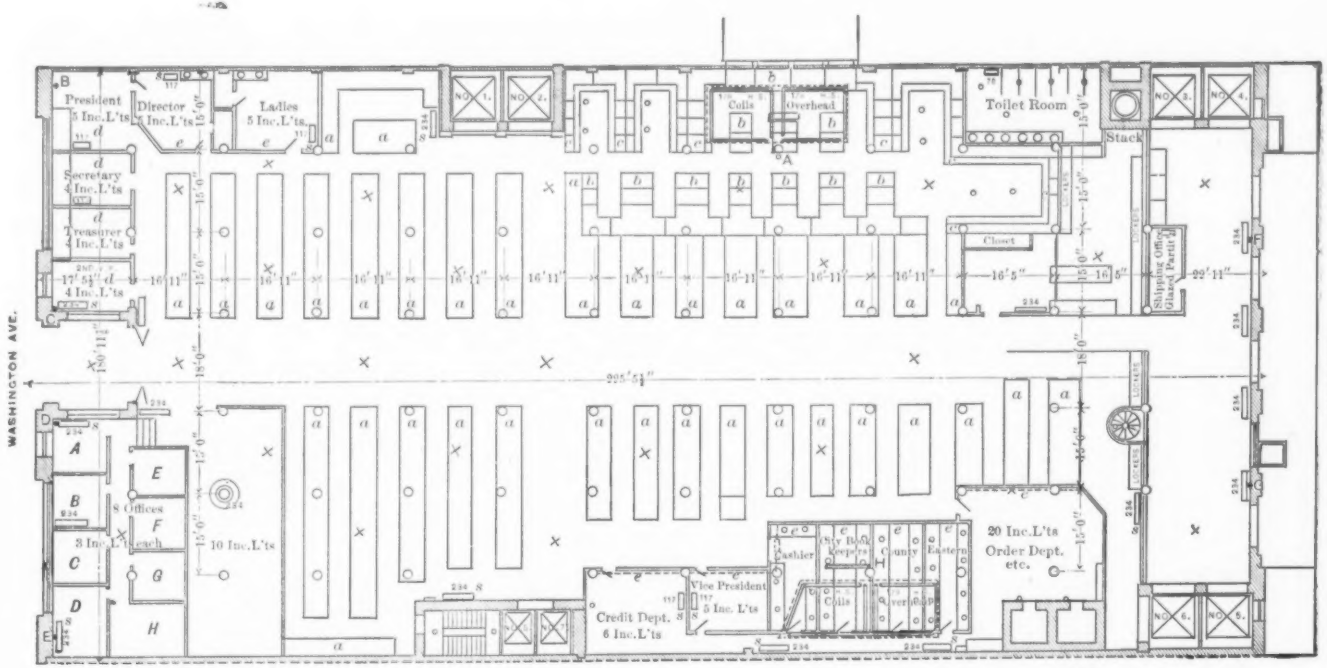

Boiler Room
Engine Room
Stack
Fan
Skid
5 Inc. L'ts.
10 Inc. L'ts.
15 Inc. L'ts.
Feeder Cables
NOTE:
INCANDESCENT LIGHTS TOTAL 48
ARC LIGHTS, " 19
HEAVY LINE DENOTES RISER RETURN
DOUBLE LINES DENOTES BASEMENT MAINS
CELLAR FLOOR PLAN
SHOWS ORIGINAL SYSTEM OF HEATING AS SPECIFIED
Bryan

FIG. 100.



a LOW BENCHES
b 3'6" COUNTER-TILLS UNDER
c HIGH FIXTURES
d LOW RAILING
e HIGH PARTITION
RISERS INDICATED BY CAPITAL LETTERS A• B• C• ETC.

W
S N
E

NOTE:
○ INCANDESCENT LIGHTS TOTAL 145
✳ ARC LIGHTS TOTAL 26
RADIATING SURFACE.
NUMBER OF COILS IN SKYLIGHT 4
NUMBER OF RADIATORS 26
b' RADIATING SURFACE IN SKYLIGHT 714
a' RADIATING SURFACE 5107
5821 TOTAL *a'*

FIRST FLOOR PLAN

Bryan

FIG. 101.

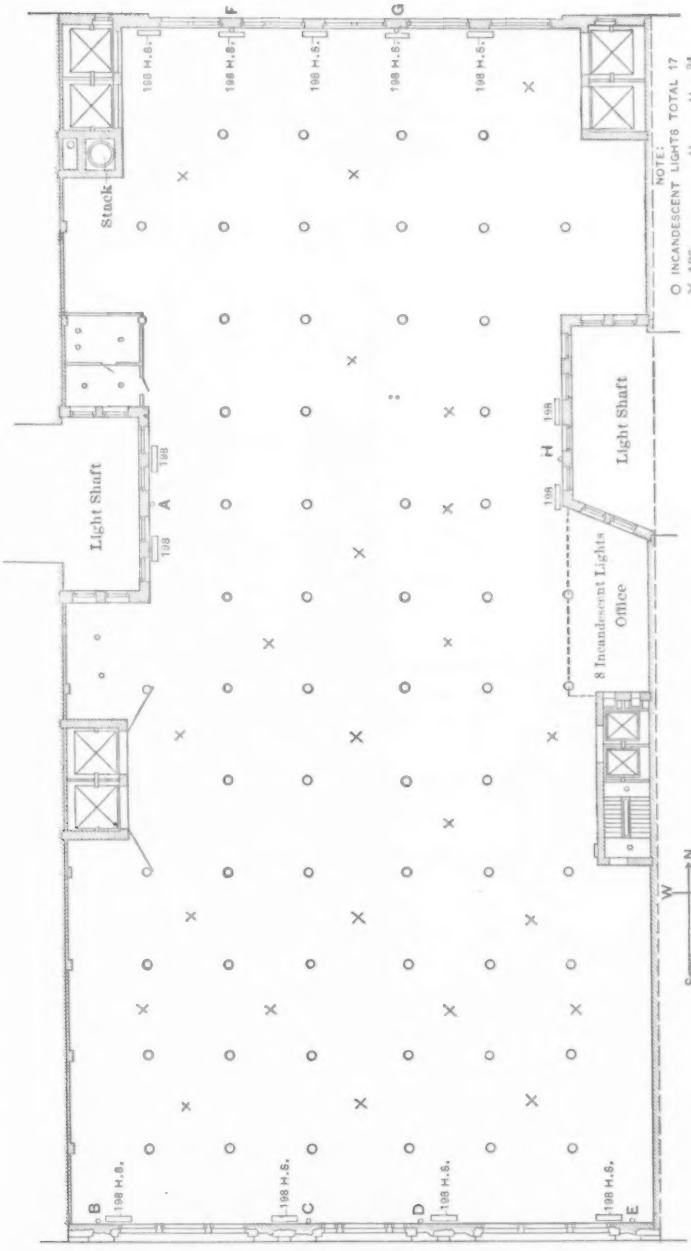
nearly as possible, detail plans and specifications were prepared, and tenders were solicited from a limited number of experienced and responsible bidders.

There were, of course, some features which could not be definitely decided upon in advance. The capacity of the boiler plant, for instance, as will be shown later, depends wholly upon the type of steam engines and elevator mechanism adopted, the most inefficient requiring nearly twice as much boiler capacity as the most efficient. There was, as usual, considerable controversy over the type of elevators to be adopted. All the modern types—electrical, hydraulic, and even steam—were warmly advocated. The size and character of the electrical plant, and arrangement of the engines and boiler room, could not be settled until these matters were disposed of.

The work of preparing detailed plans and specifications was taken actively in hand about January 10, 1898, and they were completed and sent out about February 1st, bids being asked for until noon, February 16, 1898. The specifications were, in general, in line with the suggestions made by the author in the paper presented at the Niagara meeting of the Society on "The Relations Between the Purchaser, the Engineer, and the Manufacturer" (*Transactions*, vol. xix). They embodied the following divisions:

First: The "Notice to Bidders," indicating the time and place of receiving bids; the manner of their preparation, whether on single or combined sections; each bid to be in good faith, not requiring later approval; the bids themselves to become the property of the purchaser, and not be returned. The tenders were to be made on proposal forms which were given the bidders, and not on their regular forms.

This was followed by the "General Provisions," covering the conditions under which the different divisions of the work were to be executed, such as replacing of unsatisfactory work and material; supervision; the decision of disputed points; compliance with laws, ordinances, and underwriter's rules; changes, how made; damage done to building by contractor; course to be pursued in case of failure or delay by the contractor; time of beginning and completing the work; deduction for delay in completion; services of expert; standard of excellence, where special makes of goods are mentioned by name; guarantee of one year on all parts of the work; accuracy of data furnished,



THIRD FLOOR PLAN.
CHARACTERISTIC UPPER FLOOR
FIG. 102.

Bryan

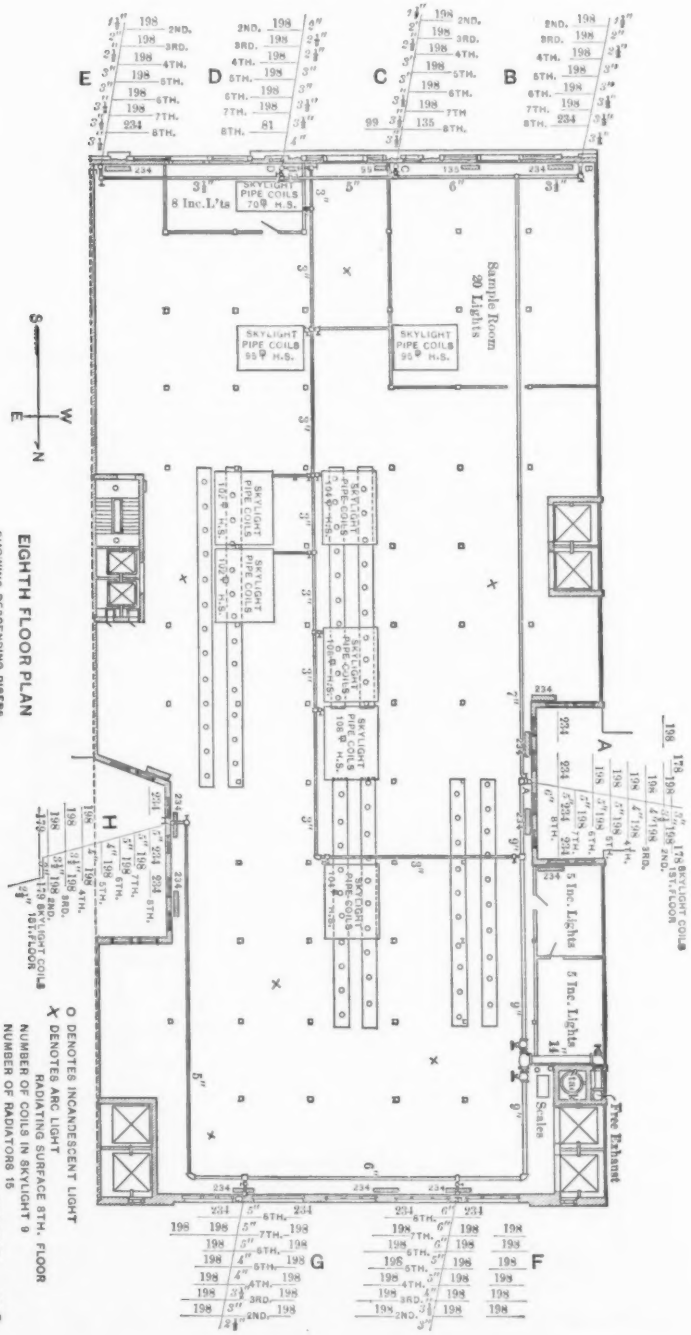
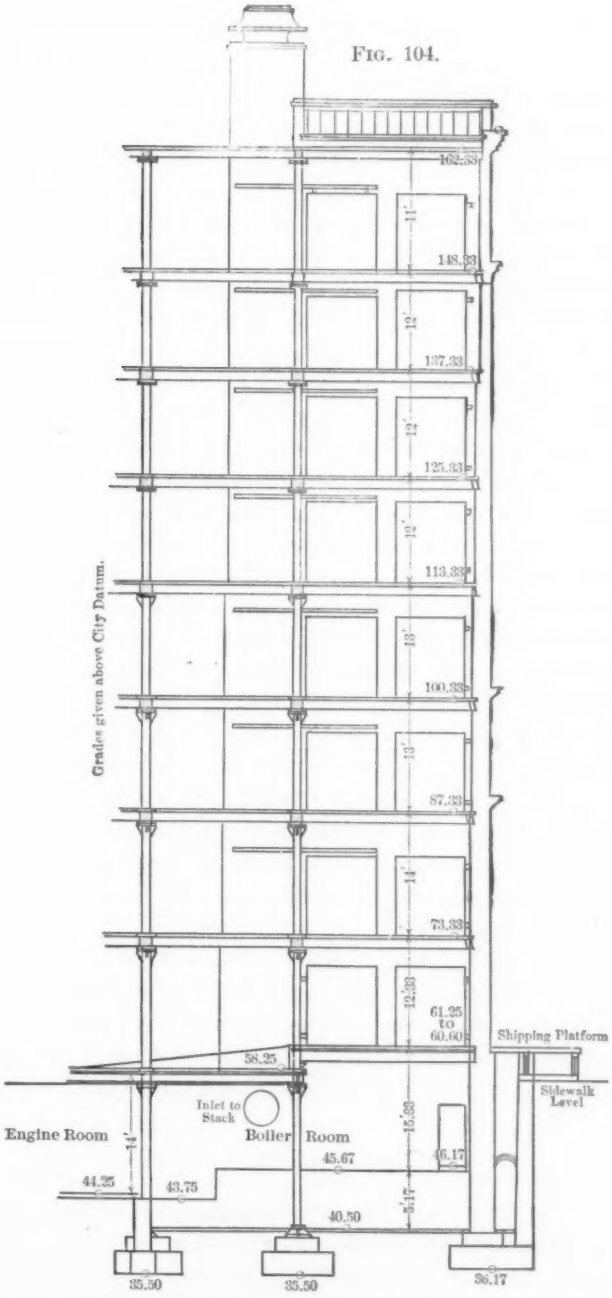

Sample Room
30 Lights
SKYLIGHT PIPE COILS 95° H.S.
8 Inc. L'ts
5 Inc. Lights
Scales
Free Exhaust
A
B
C
D
E
F
G
H
S
W
E
N
EIGHTH FLOOR PLAN
SHOWING DESCENDING RISERS.
Bryan
O DENOTES INCANDESCENT LIGHT
X DENOTES ARC LIGHT
RADIATING SURFACE 8TH. FLOOR
NUMBER OF COILS IN SKYLIGHT 9
NUMBER OF RADIATORS 15
RADIATING SURFACE IN SKYLIGHT 888
RADIATING SURFACE 3357
TOTAL RADIATION ON 8TH. FLOOR 4245

FIG. 103.

FIG. 104.



LONGITUDINAL SECTION AT REAR END OF BUILDING LOOKING WEST.

Bryan

contractor being required to verify same and see that the plant went together properly; the requirement of bond; and terms of payment. As nearly every section required foundations, a paragraph was added covering once for all the character of foundation required, the method of conducting the work, etc. The foundations, by the way, were to be built of concrete, of Atlas or Empire American Portland cement, one part of which was mixed with three parts of sand, and four to five parts of crushed macadam, cleaned and screened.

Following these come the different sections of the work itself, as follows:

Section A. The Steam-Boiler Plant, including furnaces, brickwork, smoke flue, etc.

Section B. The Elevator System; bids being invited on hydraulic, electrical, and steam, erected complete.

Section C. Dynamos and Switchboard; provision being made for additional capacity in case electric elevators were adopted.

Section D. Steam Engines for Driving the Dynamos; the specifications being written for compound engines, proposals being also invited on simple engines.

Section E. Wiring, Cables, Arc Lamps, Incandescent Lamps, Fans, and Fan Motors for ventilating.

Section F. Steam Heating System; the specifications covering a plant designed to operate under the most unfavorable conditions with very low steam pressure, proposals being also invited on the Paul and Webster Systems of air removal, with the privilege of rearranging the work to suit these systems, and reducing the amount of radiating surface, the size of mains, etc., if desired, while guaranteeing to do the same work.

Section G. Large and High-Pressure Pipework, including connections between boilers, engines, free exhaust, etc. The details of this section could not, of course, be worked out until a decision was reached as to the boilers, engines, and elevators.

The discussion of these sections will now be taken up in order:

Section A—The Boiler Plant.

Steam was required in this building for three purposes: Heating, lighting, and elevator service.

As shown later, the maximum amount of steam required for heating was about 250 boiler horse-power. Careful computa-

tions showed that more than this amount would be necessary for the lighting and elevator service, even with the most economical steam engines. As there would always be a surplus of exhaust steam available for heating, it was evident that the heating requirements need not be considerd further as a factor in determining boiler capacity.

The maximum electrical requirements are shown later herein to be about 150 kilowatts—exclusive of the elevators—or about 200 electrical horse-power. Assuming a combined efficiency of engine and dynamo unit, from steam cylinders to switchboard, of 80 per cent., the maximum indicated horse-power is 250. Using simple steam engines at a water rate of 40 pounds per indicated horse-power per hour, the steam requirements would be 10,000 pounds per hour; or, for compound engines at 28 pounds of water per indicated horse-power per hour, 7,000. These being divided by 30 to reduce them to boiler horse-power, give us 333 for the former, and 233 for the latter.

The efficiencies above assumed are not, of course, accurate, nor is the statement of the unit of boiler horse-power; but they are believed to be sufficiently close for the present purpose, in view of the impossibility of determining accurately in advance what the work will be, and the further fact that the plant would rarely be operated under conditions favorable to the highest economies. The boiler horse-power required to operate the elevators is shown later to be as follows:

For steam elevators, 67 indicated horse-power, at 100 pounds water rate, ÷ 30... 223
For hydraulic elevators, with simple duplex pumps, 75 indicated horse-power, at 100 ÷ 30 250
For hydraulic elevators, with compound steam pumps, 75 indicated horse-power, at 60 ÷ 30 150
For hydraulic elevators, with high duty pumping engines, 75 indicated horse-power, at 28 ÷ 30..... 70
For electric elevators, with dynamos driven by simple engines, 75 indicated horse-power, at 40 ÷ 30. 100
For electric elevators, if driven by compound engines, 75 indicated horse-power, at 28 ÷ 30..... 70

The following table shows some of the combinations which were possible, all of which were under serious consideration. It will be seen that the maximum steam requirements for electrical and elevator service might vary all the way between 303 and 583 boiler horse-power.

	H.P.
Compound dynamo engines and electric elevators	303
Compound dynamo engines and hydraulic elevators, driven by high duty pumping engines	303
Compound dynamo engines and hydraulic elevators, driven by ordinary duplex compound pumps	383
Simple dynamo engines and electric elevators	433
Compound dynamo engines and steam elevators	456
Simple dynamo engines and hydraulic elevators, driven by compound duplex pumps	483
Simple dynamo engines and steam elevators	556
Simple dynamo engines and hydraulic elevators, driven by simple duplex pumps	583

After due consideration, it was decided to ask for proposals on three boilers, each having a rated capacity of 6,000 pounds of water per hour, with feed water at 212 degrees into dry steam of 125 pounds pressure above atmosphere, equivalent to 181.7 horse-power each, at the Society's standard of 30 pounds from feed at 100 degrees Fahr. to steam at 70 pounds, gauge pressure. Each boiler was to be capable of doing this work continuously with natural draught under regular working conditions, and of being overworked one-third in emergencies. It was assumed that this capacity would probably take care of the building; two boilers being operated regularly, and the third being held in reserve for cleaning or repairs, or used on rare occasions of unusually large demands. There was, at that time, considerable doubt as to whether the owners could be induced to put in the most economical type of apparatus. The capacity was purposely stated in pounds of water to be evaporated per hour, rather than in horse-power, so as to avoid discussing the question of heating surface per horse-power adopted by the different makers.

The next question was as to type of boiler—whether of the water-tube or of the ordinary horizontal return tubular variety. The writer's experience had led to the conclusion that the water-tube boiler possesses three important advantages:

1*st*. Large capacity in small space.

2*d*. Large capacity in single units.

3*d*. Safety—particularly with the higher pressures required for modern steam plants of high efficiency.

On the other hand, the ordinary boiler has the advantage of lower first cost—particularly in the smaller units; simplicity, and ease of being cared for.

In the above comparison no preference is given to either type of boiler on the score of efficiency; it being the opinion of the author that there is no material difference, assuming that both boilers are designed, constructed, operated, and maintained with equal intelligence. It is undoubtedly true that water-tube boilers are actually showing better fuel economy all over the country than the average ordinary boilers, but the latter installations are usually small, and rarely receive the skilled attention in their construction and operation given the water-tube boiler.

A consideration of the requirements of the present plant clearly "indicated" the adoption of the water-tube boiler, and the specifications were drawn accordingly.

Bidders on Section A were asked to submit with their proposals the following data, in order that intelligent comparison of the bids might be made:

Make of boiler.
Number and dimensions of drums.
Number and dimensions of tubes.
Heating and grate surface.
Type of smokeless furnace offered.

The specifications for Section A covered the boilers erected in place, with all the usual fixtures and fittings, brickwork, and smoke flue leading to chimney.

The general scheme of the boiler room as originally outlined and as actually adopted is shown in Fig. 105. The boilers were designed to burn low-grade slack and nut coal from southern Illinois, and to use St. Louis hydrant water. The boilers and furnaces were to have a combined efficiency of at least 70 per cent. of the calorific value of the fuel when operating at anywhere between their rated capacity and 20 per cent. above. The evaporative trials to be made with a coal equivalent to Mount Olive lump, having a calorific value of not less than 11,000 British thermal units per pound. The entrainment was not to exceed 1 per cent. when operated at rating, or 1½ per cent. when operated at one-third above rating. The working pressure was to be 125 pounds, with a factor of safety of 5. The boilers were to be submitted to a hydrostatic pressure test of 50 per cent. greater.

Material in shells and heads was to conform to the latest

specifications of the American Boiler Manufacturers' Association, as follows: Steel to be open-hearth flange or fire-box. Tensile strength, 55,000 to 62,000 pounds per square inch; elastic limit, one-half the ultimate; elongation, at least 30 per cent. in 8 inches; maximum phosphorus, .03 per cent.; maximum sulphur, .025 per cent. Material to stand bending double on

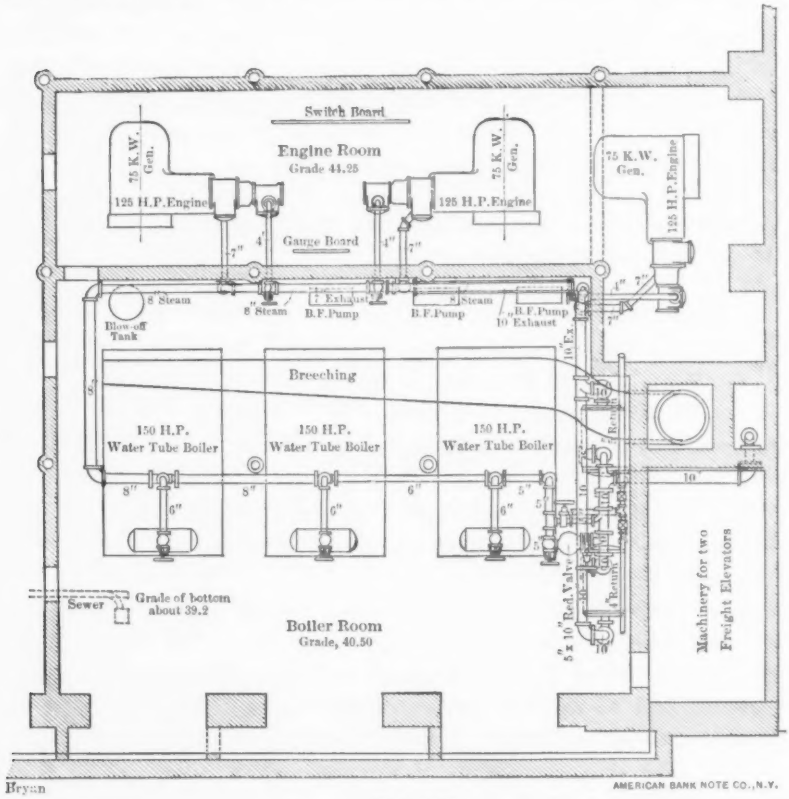


FIG. 105.

itself, both cold and after being quenched in water of 70 degrees Fahr. temperature. The tubes were to be 3½ or 4 inches external diameter, of best charcoal iron or soft steel, well expanded into tapering holes, which were either to be drilled out or punched and reamed. The rivets were to be capable of bending double on themselves, both cold and after being heated and quenched. Rivet holes were not to be more than one-sixteenth larger than rivets, and to be punched one-eighth inch smaller

than rivets, and then reamed out full size. The improper use of the drift pin was prohibited. The other detail requirements of the specifications did not involve anything special, except that a stairway was to be provided, affording ready access to the tops of boilers. The contractor was to look after the city and insurance tests, and turn the boilers over ready for opera-

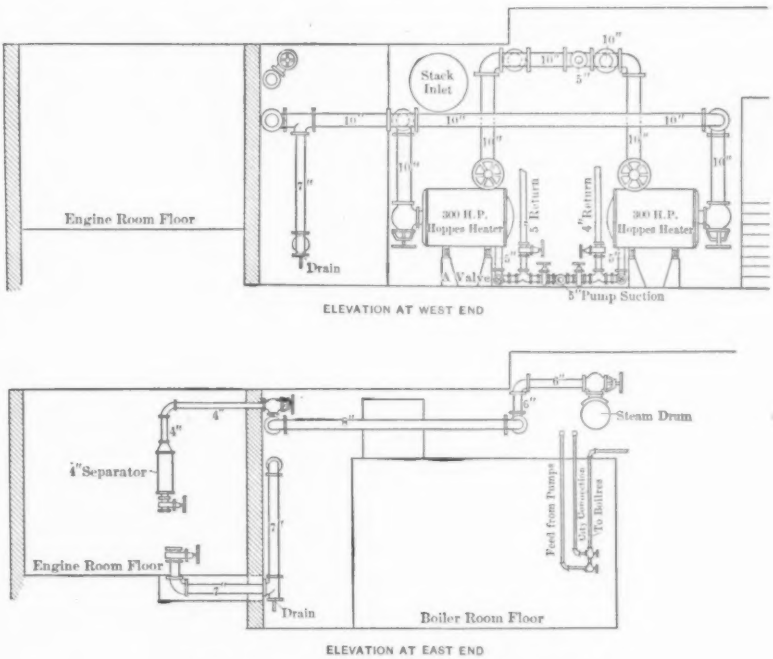


FIG. 106.

tion. Dampers were to be provided so as to be readily controlled and adjusted from the front of boilers.

The contractor was to furnish and attach to each boiler a smokeless furnace, equivalent in capacity, efficiency, and smokelessness to the down-draught with lower grate. This requirement was inserted for two reasons: First, this type of furnace unquestionably improves the fuel efficiency of the boilers; and, second, the plant being located in the heart of the city, it was desired to avoid any smoke nuisance, although the city smoke

ordinance has been declared invalid by the Supreme Court of the State.

The writer specified the same form of fire doors which he first used in the summer of 1897, opening across the entire width of the grate surface, permitting access to all parts of it for observation, slicing, and cleaning. This has proved a very desirable feature in practice, as the upper doors of down-draught furnaces usually stand open, and a central division or column is very much in the way.

Very few smokeless furnaces have been successful with the low-grade fuels burned in St. Louis. The automatic stokers, firebrick arches, and steam jets have met with limited success under favorable conditions, but have not proved themselves adapted for excessive and fluctuating loads, nor have they demonstrated any large fuel economy. The down-draught furnace, therefore, appears to be the only device which can be recommended with confidence, and even it is not always available, on account of its first cost, and its liability to injury if ignorantly or carelessly handled.

The chimney (Fig. 104) was included in the building contract. It was located inside a square brick shaft, and was made of riveted steel $\frac{1}{4}$ inch thick, resting on a substantial cast-iron foot plate. It was lined its entire length with firebrick. Its dimensions were 48 inches internal diameter by about 135 feet high, above floor level of boiler room. It would ordinarily be rated at about 400 horse-power. The dimensions were decided upon when the building was begun, before any of the computations as to actual steam requirements had been made. It was assumed that only two boilers would ever be operated at one time, and that the stack could be overworked in emergencies, as experience has shown to be possible.

Section Aa—Horizontal Return Tubular Boilers.

Thinking that the cost of the water-tube boilers might be greater than the owners cared to invest, it was decided to ask for alternate proposals on horizontal return tubular boilers of as nearly the same capacity as possible. The plant was divided into four units, three of which were to be able to do as much work as two of the water-tube boilers. The size best suited to the space available was 66 inches in diameter by 20 feet long,

with sixty-four 4-inch tubes 20 feet long. At 12 square feet heating surface per horse-power, these boilers would be rated at 131 horse-power each, or 393 for three, and 524 for the entire plant.

The chief advantage of boilers of this type is supposed to be lower first cost, although this disappears, to some extent, on large boilers built for high pressures. The belief is common that such boilers cost less for repairs, and are more easily operated and cleaned, than the water-tube boiler. While these claims have been proven in some cases, the opposite has also been shown to be true in other instances, so that the truth of either proposition cannot be said to have been established as yet.

The ordinary boilers designed for this plant are shown and briefly described in the author's discussion of Mr. Woolson's paper on "The Hanging and Setting of the Horizontal Fire-Tube Boiler," read at the Niagara meeting of the Society (vol. xix., *Transactions*). The design embodies what the writer believes to be many excellent features, together with proportions which are intended to insure the highest possible fuel efficiency obtainable with low-grade fuels which come to the St. Louis market. The general features of the water-tube specifications were preserved, except that no guarantee was required as to capacity, efficiency, or entrainment, as it would have been unfair to have held the manufacturer responsible for a design in which he had no part. The fittings and general construction were high grade in every respect, so as to be strictly comparable with water-tube work.

Section B—The Elevator System.

This division presented the most complicated problem encountered, and one on which there was the widest difference of opinion among interested parties. Steam elevators had been used by the tenants in the old building with entire satisfaction. The architects had recently installed in a large system of warehouses a modern high-pressure hydraulic plant, which had displaced a system of steam elevators, and had made an enormous fuel saving. The engineers had devoted considerable time to the study of modern electrical elevators, and recognized that they were destined to come into extensive use. They were

not, however, fully convinced that this was the proper system to adopt in the present instance, and were fully aware of the advantages of the other two types. It was decided, therefore, to so word the specifications that comparative bids would be received on all three types.

Some interesting data were secured from the tenant's old plant. Tests were made on their single passenger and three freight elevators on an average busy day. It was found that the passenger elevator was in actual motion 47 per cent. of the time, and the freight elevators, 35 per cent. of the time. To determine the loading of the passenger elevators, the 6 foot by 6 foot car was filled with people who happened to be conducting business in the store at the time. Eighteen were crowded into the car. Estimating their weight at 2,700 pounds, the loading was 75 pounds per square foot, which would seem to be a safe figure for passenger cars.

The tenants indicated their elevator requirements as follows (see Fig. 101): There were to be two passenger elevators (Nos. 7 and 8), located in the southeast corner of the building, each to have a safe carrying capacity of 2,500 pounds of live load at a maximum speed of 300 feet per minute. These elevators were to be operated by conductors and to run between basement and top floor, stopping at all floors. The other six elevators were to carry freight only. The two southwest elevators (Nos. 1 and 2) were to be operated by conductors, and to have a capacity of 3,500 pounds at a speed of 225 feet, operating from basement to top floor, stopping at all floors. Their duties were to carry sold goods up to the top floor to be packed for shipment. The northwest (Nos. 3 and 4) and northeast elevators (Nos. 5 and 6) were to carry 4,000 pounds at 150 feet. The northeast elevators were to be operated by porters, to carry packed goods in both directions, stopping at all floors. The northwest elevators were to carry packed goods from the eighth floor to shipping room. They did not run to the basement, nor stop at intermediate floors, and were operated without attendants on the cars. The dumbwaiter was to be of the electrical type, designed to carry 200 pounds 100 feet per minute. The contractor was to provide ornamental metallic gates on each floor for passenger elevator shafts. Separate proposals were invited for safety gates at the freight elevator doorways on all floors.

The original specifications were written to cover hydraulic

elevators, alternative proposals being invited on electric and steam machines capable of doing the same work. Contractor was to furnish the machinery complete ready for service, and allow $300 for each passenger car. The travel of the northwest elevator was about 87 feet, and that of all others 105 feet. Working steam pressure, 120 pounds. The water pressure for the hydraulics was not to exceed 750 pounds at the pumps. Guide posts, of steel. Cables to be of such number, dimensions, and strength that in case half of them should fail simultaneously, the remaining half would carry the maximum load, with a factor of safety of 4. Counterbalancing was to be employed as far as possible. Complete safety devices were to be supplied, preventing excessive speeds in either direction; automatic stops at end of travel, and other features, such as slack cable stops, buffers, grips, etc. The pumping engines were to be two in number, each sufficiently large to handle the plant at a piston speed not exceeding 100 feet per minute, with the aid of the accumulator. Both pumps were to be of the horizontal direct-acting duplex type with compound steam cylinders, and outside packed water plungers. One was to be of the high duty pattern, having a water rate of not over 28 pounds per indicated horse-power hour. The other pump was to be of the ordinary duplex pattern, with a water rate not exceeding 60 pounds per indicated horse-power hour. There was to be a weighted mechanical accumulator for the hydraulic system of such capacity that with six elevators operating continuously in the same direction, stopping only at the top and bottom floors, and one pump in operation at not exceeding 100 feet piston speed, the elevators would do the specified service. The discharge tank was to be located in the attic. Contractor was to furnish and erect all necessary apparatus, and do all water piping. He was to furnish the lubricating apparatus, paint the plant, and submit it to such tests as the engineers might require.

The determining of the horse-power required to operate the elevator plant presented many difficulties. There was at that time—and still is—a lack of authentic data as to elevator performance covering wide ranges of service, and it is still impossible to make such computations accurately. Extreme accuracy, however, is not necessary, in view of the fact that it is impossible to predict what the average loads and percentages of time in operation will actually be in a prospective plant.

The problem was finally solved for hydraulic elevators in the following manner: The average net horse-power required by each freight elevator lifting its full live load was 20 horse-power, and for each passenger elevator 22½ horse-power. There were six of the former in operation one-third of the time, but as they use no power coming down, the average work done was 20 net horse-power. The two passenger elevators were assumed to be in operation one-half of the time, and their requirements were, therefore, 11¼; a total for the eight machines of 31¼ net average horse-power. Assuming an average efficiency of 55 per cent. from water end of the pump to work done, the horse-power of water end of the pump would be 56. On the basis of a mechanical efficiency of 75 per cent. for the pump, the indicated horse-power of the steam end is 75, the figure already used in determining the boiler capacity.

The electric elevators, however, presented greater complications. Tests have shown that where drum machines are counterbalanced to half the average load, and where the same total load is carried both up and down, the net work done is simply that necessary to overcome friction, plus an allowance to cover the "surge" of energy required at each start. A study of this subject, in connection with records of tests made on similar plants, indicated that when operated under favorable conditions of loads, counterbalancing, frequency of stops, etc., these elevators should have an efficiency, running in both directions, about as follows:

The two passenger elevators	3½	kilowatts	hours	per	car	mile.
The two southwest freight elevators	4¼	"	"	"	"	"
The four north freight elevators	4½	"	"	"	"	"

On this basis the electrical-horse power required to keep each elevator in motion was found to be 4⅔, 5⅔, and 6, respectively, per car mile. The speeds specified, however, were 3.4, 2.55, and 1.7 miles per hour, respectively, on which basis 15.9, 14.45, and 10.2 electrical horse-power were required. Assuming that the passenger elevators might be in motion one-half the time and the freights one-third, and multiplying by the number of elevators, the power consumed is found to be 15.9, 9.6, and 13.6 electrical horse-power, respectively; a total of 40, in round numbers. In view of the fact, however, that ideal conditions of loading and counterbalancing would never be reached, and also

that uncertainty existed as to the actual number of stops per trip, and percentage of time which the elevators would be in operation, and in order to avoid any appearance of partiality to the electric installation, it was thought wise to increase this allowance by 50 per cent., and to call the total 60 electrical horse-power at switchboard. Assuming a combined engine and dynamo efficiency of 80 per cent., the average indicated horse-power at cylinders of dynamo engines is 75, as already stated.

In computing the horse-power required to drive steam elevators, it was presumed that they would be run by direct-connected steam engines in the ordinary manner. The machines are almost identical with the electric elevator of the drum type operated by worm gear, the direct-connected steam engine taking the place of the distant steam engine generating current through a dynamo, and transmitting it for use in a direct-connected electrical motor. The work done by the elevator proper is the same in both cases. Assuming the efficiency of the motor on the electric elevator to be 83 per cent., the net power consumed by the elevator mechanism itself is 50 horse-power. Taking the mechanical efficiency of the steam elevator engines at 75 per cent., we arrive at the figure of 67 above referred to.

The clause in the specifications providing for alternative bids on electrical elevators stipulated that the elevators were to be of the same general character as the hydraulic, modifying the design only in so far as necessitated by electrical driving. The purchaser agreed to supply electrical current at 220 volts at the switchboard. The contractor was to do all wiring from that point to elevators. Elevators to be of the single or double worm type, with steel worm and gun-metal gear. The motor was to have an automatic starting device to give the car an easy start independent of the operator and cable, returning to the starting point automatically in case of interruption of the current. When starting from rest to full speed within five seconds, the starting current was not to exceed the operating current at full speed by more than 50 per cent. The purchaser was to provide a third electrical generating unit of 75 kilowatts capacity in case electric elevators were adopted. The plant to be so designed as to be capable of operating in multiple with the lighting system without noticeable interference with the steadiness of the electric lights.

The alternate proposals asked for on steam elevators were to

comply with the same general conditions as before, except that the elevators were to be driven by direct-connected steam engines of the duplex double-acting vertical type. Separate proposals were invited on an ash hoist to have a platform of about 60 inches by 33 inches; capacity, 1,000 pounds; speed, 50 feet; travel, about 20 feet.

Section C—Dynamos and Switchboard.

The contractor for Section C was to furnish and erect the generators and switchboard, with connections between same, as located in Fig. 109. Proposals were to state the commercial efficiency of the generators at $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, and full load, subject to test after the machines were installed. A study of the purchaser's needs showed that they could be best met by a combination of arc and incandescent lamps. A preliminary location of arc and incandescent lamps was agreed upon, showing approximately 200 arc and 600 incandescent lamps. In addition, there were three motors driving fans in basement, and an allowance was also made for small fan motors throughout the house, and about 1 per cent. for drop in the wiring. These were summarized as follows:

200 arcs at $2\frac{1}{2}$ amperes, and 220 volts, or 550 watts.....	= 110,000	watts.
600 —220-volt 16 candle-power incandescents at 60 watts... ...	= 36,000	"
For motors, drop, etc.....	4,000	"
Total.....	150,000	"

It was decided that this load could be best handled by two 75-kilowatt generators. As such machines can run at considerable overloads, two-thirds of the total lights could be handled with one machine out of service. Three 50-kilowatt machines would have been better, but would have cost more.

The selection of dynamos whose rated capacity was equivalent to the total number of lights connected, may be criticized, in view of the fact that ordinarily the maximum number of lights burned at one time—even for short intervals—is less than the number connected, and that the generators can be overworked considerably for short periods. The anticipated load factor, however, was large, and some margin for growth was necessary. No special or reserve machine being provided, it was not desired to cripple the plant more than absolutely neces-

sary when one machine might be shut down on account of accident or for repairs.

One hundred and fifty kilowatts at the switchboard is equivalent to 200 electrical horse-power. Assuming a combined efficiency of engine and dynamo unit of 80 per cent., the indicated horse-power in the steam cylinders is 250, as already stated in estimating boiler capacity.

The question might be asked why the maximum load was taken for the boiler horse-power, when the average load would be much less, and the entire number of lights would probably never be burned at one time. As explained above, the work done in this building was of such a nature that what is known as the "load factor," or ratio between the maximum number of lights operated at one time and the total number connected, would be large, and it would sometimes happen that every light might be burning. Furthermore, experience has shown that more lights and motors are added as business increases and the necessity for them develops, so that some margin for growth must be allowed.

A departure from established practice was made in adopting a voltage of 220, instead of 110. This was done for three reasons :

First, it was desirable to have an outside connection from a central station for use in the event of any serious accident to the plant. No central station, however, furnished direct current for power at 110 volts, but two convenient stations offered such service at 220 volts, both being from 220–440 direct-current three-wire systems.

Second, it was believed that the electric motors would give better satisfaction at this voltage, and stopping and starting them would have less effect on the lights, as it was intended to run the entire light and power service from the same electrical apparatus.

Third, the distances covered being quite large for an isolated plant, both the investment in copper and the "drop," or percentage of loss in wiring system, were materially reduced.

It is true that the 220-volt incandescent lamp is not as efficient as the 110, and that it costs more, thus increasing the cost both of fuel and of lamp renewals. The 110-volt 16 candle-power lamp ordinarily used consumes about 55 watts, and the 220 volt about 60. This means 9 per cent. more generating capacity

= 120,000 watts.
= 30,000 "
4,000 "
150,000 "

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and fuel burned. The 16 candle-power 220-volt lamp costs from 23 cents to 25 cents each, and the 110-volt 18 cents to 20 cents. This inferiority is largely due to the fact that the number of 220-volt lamps in use is still small. The efficiency is improving, and the price falling, as the demand increases, and it is believed they will, in the not distant future, approximate closely to the 110-volt lamp. In the present case it was believed that the advantages of the 220-volt system overbalanced the objections named.

At the time this work was taken up the 220-volt arc lamp was in an experimental stage. It was a question whether to burn the arcs singly across 220 volts, or to use two 110-volt lamps in series. In either case, the enclosed arc lamp, burning in multiple, was to be used, for reasons explained later. A large number of sample 220-volt arc lamps were tested in the offices of the engineers, and a reasonable number of these were found satisfactory. The arc is long, and the light has a bluish purple tinge. The former, however, was not objectionable with the ground-glass inner globe. The latter also yields to treatment; so that, on the whole, it was thought entirely safe to use the single lamp, thus greatly simplifying the system.

The electric elevators could of course, have been operated alone with less than 75-kilowatt average; 75 horse-power, or 56 kilowatts, being the average amount of energy required, as already explained. As electric elevator service, however, is frequently subject to sudden and severe overloads for short periods, due to the starting and operating of a number of loaded elevators simultaneously, a surplus of power was necessary; and as it was desirable to have all the electrical units alike and interchangeable, it was thought best not to reduce the size of the third unit.

The specifications provided that the generators should be of the direct-connected type, mounted on engines, each generator to be rated at 75 kilowatts at 220 volts. Speed about 250 revolutions per minute. When run continuously at rated load, the temperature was not to rise more than the following amounts above the surrounding air: 72 degrees Fahr. in the armature; 54 degrees in the fields; 72 degrees in the commutator. They were to be capable of carrying 25 per cent. overload for two hours, and $33\frac{1}{3}$ for one hour, without excessive heating anywhere, and without injurious sparking at the brushes. They

were to be capable of taking 50 per cent. overload momentarily, without flashing or injurious sparking, and were not to flash around the commutator when the circuit breaker opened at 50 per cent. overload. They were to run without sparking or shifting of the brushes from no load to 25 per cent. overload.

These rather rigid requirements were due to the possible operation of electric elevators, and as several prominent builders were in a position to supply such machines, competition was not thereby limited.

The winding of the fields was over-compounded to give 3 per cent. higher voltage at the brushes at full load than at one-tenth load. The rise in voltage was to be proportional to the load throughout. The current density at rating in the carbon brushes was not to exceed 27 amperes per square inch of brush contact. The insulation resistance between any conductor and the frame of dynamo was to be at least one megohm.

The switchboard (Fig. 109) was to be of white Italian marble, 1½ inches thick, located 24 inches from wall. There was to be a generating panel for each dynamo, containing one fire-proof field rheostat, one double-pole double-throw knife switch, one single-pole magnetic circuit breaker, one pilot lamp, one Weston ampere metre, the necessary bus bars, one round pattern Weston 250-volt voltmeter, with switches to connect it with the terminals of each dynamo; with bus bars, with the distributing circuits, with positive bus bar and ground, and with negative bus bar and ground. There were to be two feeder panels, containing one 1,200-ampere double-pole double-throw switch, ten 150 and four 50-ampere double-pole double-throw knife switches with fuses, and one lamp ground detector. Each feeder panel was to have a neat goose-neck bracket light, with shade and holder. The conductors were to be proportioned so that the temperature would never rise more than 45 degrees Fahr. above the surrounding air. Each circuit on the board was to be lettered or numbered.

This board is designed with an extra power bus bar, so that the elevators can be operated separately from the lights, if desired. This has never been done, however, except when the plant was started, and before engines and generators were adjusted. The large double-pole double-throw switch enables either the motors or lights to be connected to outside reserve connection, the other being operated from the dynamo. When

both sides of this switch are thrown to outside connections, the entire building is operated from central station on three wire 220-440 system.

Each generator was to be connected with switchboard by rubber-covered cable, running overhead on porcelain knobs. Each proposal was to name a price on two generators as above, with an extra price for a third unit, with one feeder panel, the latter to contain eight 150-ampere double-pole single-throw switches; eight 200-ampere single-pole circuit breakers, and one 220-volt 500-ampere recording station wattmeter—all of these being required for the electric elevator service.

Section D—Steam Engines.

These were to be of the direct-connected type, for driving the two 75-kilowatt generators, each to be operated independently (see Fig. 105). The assumed efficiency of 80 per cent. fixed their capacity at 125 indicated horse-power each. The operating steam pressure, if compound engines were selected, was to be 120 pounds above atmosphere. Contractor was to furnish and erect these engines, ready for pipe connections, including foundations for engines and dynamos. Each proposal was to embody a guaranteed water rate per indicated horse-power hour, and be accompanied by a drawing showing space occupied, foundations required, dimensions of cylinders, weight of engine, fly-wheel, and sub-base, and diameter of shaft. The proposal was also to state additional cost of a third unit identical with the others, for use in case electrical elevators were adopted.

The specifications were drawn for compound engines. Condensing apparatus was not deemed desirable, on account of its increased cost, complication, and space required. The fuel saving would have been small, as the exhaust is used for heating during the five months of cold weather, when the load on the engines is also a maximum. Bidders were asked to state the price of simple engines of the same indicated horse-power. The engines were to be of the horizontal centre-crank, tandem compound, high-speed, non-condensing automatic type, built extra heavy throughout, speed about 250, stroke at least 14 inches, designed for an overload of 50 per cent. for short periods; water rate not over 25 pounds per indicated horse-power hour; shaft governor to be equivalent to the Rites, controlling both

high and low-pressure cylinders. The drop in speed with constant steam pressure between no load and 50 per cent. overload was not to exceed 2 per cent., and was not to exceed 1½ per cent. with constant load and steam fluctuating between 100 and 125 pounds; and 3 per cent. for the combined changes in load and pressure. Engines were to operate smoothly, noiselessly, and without heating or undue wear, at all loads and pressures. Heavy cast-iron sub-bases were to be furnished of proper width to receive dynamos. Shaft to be of steel, in one piece. The usual fittings, fixtures, and lubricating apparatus to be furnished. One extra heavy fly-wheel, steam cylinders lagged, bright work highly finished. Crank pin to be of same dimensions as shaft. Engines to be finished, painted, and submitted to such tests as might be required.

Section E—Wiring, Arc and Incandescent Lamps, Fans, and Motors.

This section covers all the apparatus and material necessary to transmit the electricity from the switchboard to the points of use. The 220-volt multiple arc distribution was adopted, for reasons already explained. The arc and incandescent lamps were run from the switchboard on independent systems of wiring. Suitable cut-out cabinets, tablet boards, and switches were to be supplied. The feeders for arc lights were to be of No. 2 B. & S. gauge, on porcelain knobs in the basement, and on metal cleats in the elevator shaft. All feeder wires were to have a one-sixteenth lead cover in elevator shaft, and were to be standard and continuous, without joints, except where taps were taken off feeding-tablet boards on each floor. All wiring was to be rubber covered. The distributing circuits for arc lamps were of No. 12 B. & S. wire, each two lamps being controlled separately. The incandescents were to be grouped in circuits not exceeding fifteen lamps each. Maximum drop between cut-out cabinet and lamps, ¾ per cent.; no wire smaller than No. 14 B. & S. to be used. Contractor was to furnish and attach lamps and sockets, the general plan of wiring and location of lamps being shown in Figs. 100, 101, 102, and 103. On the upper floors the incandescent lamps were to drop from the ceiling by flexible cords with ceiling cut-outs and key sockets. The paragraph referring to insulation was as follows:

"The insulation of all wires in this building must be of best

Para rubber, put on in two coats of uniform thickness without seams, the inner coat to be free of sulphur or other substances liable to corrode the copper. The copper to have an additional protection of a uniform and thick coating of tin. Thickness of insulation of the No. 14 and No. 12 wire to be $\frac{3}{64}$ of an inch, upon No. 2 B. & S. wire $\frac{4}{64}$ of an inch, and upon No. 000 wire $\frac{5}{32}$ of an inch. The rubber to be covered with a substantial braid that will not readily carry fire. This wire must show an insulation resistance of at least one megohm per mile after two weeks' submersion in water at 72 degrees Fahr., and after three minutes' electrification with a current of 500 volts. The insulation of all of these wires, with all cut-outs, switches, and fuses in place, must have a resistance between conductors, and between all conductors and the ground, of not less than $\frac{1}{2}$ megohm per mile of circuit."

Edison key sockets were to be used throughout, with porcelain bases. Contractor was to furnish and connect six hundred 16 candle-power lamps, with anchored filaments, and 600 hours guaranteed life. The intention was to select that type of lamp which would give this life with the best guaranteed efficiency and maintenance of candle power.

Three motors for ventilating fans were to be supplied—one in basement, one in engine room, and one in boiler room (see Fig. 100); the air to discharge into space surrounding stack. Each fan was to be 24 inches in diameter, with speed variable between 200 and 600 revolutions per minute. Motors to be $\frac{1}{2}$ horse-power each, series wound, equipped with switches, regulating devices, iron-clad armatures, large mica insulated commutators, carbon brushes, and self-oiling bearings.

The two hundred arc lamps were to be of the enclosed type, burning 150 hours continuously without trimming. Proposals were invited on lamps to burn singly on 220-volt circuits consuming not over $2\frac{1}{2}$ amperes each, and also upon lamps to operate two in series, without robbing each other and with equal steadiness. All lamps were to have resistance enclosed in top of lamps, and be hung from ceiling on independent supporting wires. They were to have opalescent inside globes, and clear glass outside globes. To have the usual fittings and adjustments, and polished brass finish. Thirty days' supply of carbons was also to be furnished.

The enclosed arc lamp has many important advantages, among

which are—that it can be run off of the ordinary incandescent, dynamo, and wiring system, and that it gives a very uniform and attractive light. It does not give as much light as the ordinary open-series arc, but requires more energy, about one-third of which goes to waste in the resistance in the lamp which is necessary to control and regulate its operation. The 110-volt enclosed arc lamp, for instance, is operated with 5 amperes, but only about 80 volts are consumed in the arc itself. The 220-volt lamp uses 2½ amperes, consuming about 160 volts in the arc. Thus far all attempts to do away with this wasteful resistance have failed, but it is believed that the inventive genius of the country will, in the near future, overcome this serious difficulty.

The original schedule of lamps to be wired for was as follows:

	Arc.	Incandescent.
Basement	19	46
1st Floor	26	145
2d "	23	15
3d "	24	17
4th "	24	39
5th "	22	29
6th "	23	46
7th "	22	100
8th "	6	142
Total	189	579

Section F—Steam Heating.

In installations of this character, all the pipe work—both high and low pressure—together with necessary apparatus, is usually included in one specification and contract. In the present instance, however, the details of the high-pressure and exhaust connections and apparatus could not be determined upon until the boilers, engines, and elevators had been selected. The heating plant was therefore taken up as an independent problem. As shown in Fig. 99, the building, though large, was favorably situatedas regards heat losses, being protected by the walls of adjoining buildings on both east and west sides, except at the two small light shafts shown (Figs. 102 and 103). The rear exposure is directly to the north, and is the only unfavorable one.

After due deliberation, it was decided that the most severe conditions would be met by assuming an external temperature of 110 degrees Fahr., the building to be heated to 70 degrees

Fahr., with steam pressure not exceeding 2 pounds. The temperature occasionally drops below —10 degrees in St. Louis, but on these rare occasions it was thought that the pressure could be carried a little above 2 pounds, or that a slightly lower temperature than 70 degrees might be permissible. The heat loss was computed on the basis of 80 British thermal units per hour per square foot for exposed glass surface, and 16 for brick wall—the units given by Wolff. A serious problem was to determine what heat loss there would be from the east and west walls, which were in contact with adjoining buildings. It was finally concluded to assume that one of these buildings might be vacant in cold weather, and might reach as low a temperature as 32 degrees Fahr. On this basis the heat loss would be about 10 British thermal units per hour per square foot of wall. The changes of air to be provided for were assumed at 4 for the shipping room on first floor, 2 for the balance of first floor, and 1 for the rest of building.

From Carpenter we learn that 1 British thermal unit will heat 55 cubic feet of air 1 degree, and following his reasoning we get the following formula for total heat loss:

$$H = 80G + 16W + \frac{80nC}{55} + 10w.$$

in which:

G = exposed glass surface in square feet.
W = exposed wall surface in square feet.
w = wall surface adjoining west store.
C = cubic feet of space to be heated.
n = changes of air per hour.

This formula gives the amount of heat which the system must supply to replace the heat lost from wall and glass exposure when the building is maintained at 70 degrees Fahr., and the external temperature is —10. It also provides heat to bring the fresh air entering building through doors, windows, etc., to the same temperature.

It has been shown that one square foot of ordinary direct radiating surface will give off about 280 British thermal units per hour, with steam of 2 pounds' gauge pressure. Dividing the heat loss by 280, and reducing, we get the following formula for radiating surface:

$$R = \tfrac{3}{7}\left(G + \frac{W}{5} + \frac{w}{8} + \frac{nC}{55}\right).$$

In using this formula, the factor $\frac{2}{7}$ was changed to $\frac{1}{3}$ to allow for the severe northern exposure, and to admit of the building being heated quickly when cold.

No special ventilation was provided for the upper floors. The rooms were all very large and occupied by but few people. It was thought, therefore, that sufficient fresh air would find its way through the doors, windows, stairways, elevator shafts, etc., to keep the rooms pure. As this air would have to be heated, however, it was, as already stated, assumed that the leakage would amount to two entire changes of air on main floor, four in shipping room—where the doors were constantly opening and closing—and one on the upper floors. The roof not being ceiled in, was assumed to have a heat loss equal to that of the exposed brick walls. This, with the skylights, accounts for the large amount of heating surface on the eighth floor. On this basis the radiating surface was found to be as follows:

Floor.	Location.	Number of Coils or Radiators.	Square Feet Radiation.	
			Specified.	Paul.
1st	Skylights	4— 714 sq. ft.		
1st	Store	21—4,054 "		
1st	Shipping room	5—1,053 "	5,821	3,372
2d		14	2,772	2,184
3d		13	2,574	2,105
4th		13	2,574	2,054
5th		12	2,376	1,944
6th		12	2,376	1,944
7th		11	2,178	1,716
8th	Store	16—3,357 "		
8th	Skylights	9— 888 "	4,245	2,754
	Total	130	24,916	18,073

The last column shows the radiating surface put in under the Paul system, and guaranteed to do the same work.

The heat loss in basement being very small, no radiating surface was provided here. The steam and return pipes, however, were all left uncovered.

The general arrangement and location of radiators, mains, and branches are shown in Figs. 100, 101, 102, 103, 105, and 106. The system of steam distribution was that of a main distributing riser going to the top floor, and there branching both ways (see Fig. 103) along the four walls, and supplying single-

pipe descending risers, from which branch connections were taken to all radiators. A few coils of 1-inch pipe were placed in skylights and provided with double-pipe connections. The basement return for descending risers was divided into two sections (Fig. 100). The radiators on first floor were to be

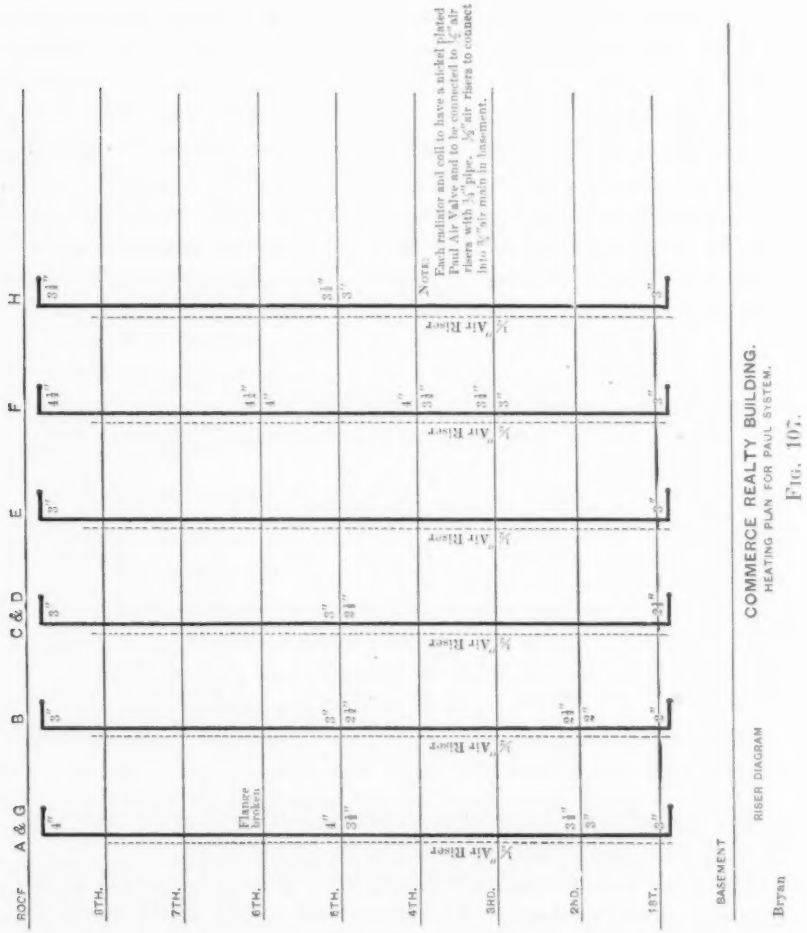


FIG. 107.

supplied by an independent single-pipe main, going entirely round basement, and carrying both steam and returns.

The selection of the maximum heating pressure at between two and three pounds fixed the velocity of steam in mains at about 37½ feet per second, causing a drop of pressure about .1

pound in 100 feet. These values were used in proportioning the mains and branches with proper allowance where the double-pipe system was used.

It was recognized that the unit values here assumed would result in a liberally designed system, capable of heating and circulating easily and quickly, and of taking care of extreme conditions satisfactorily. Many engineers would have figured the work more closely, and have taken the chances on satisfactory service, but it was not deemed good engineering to do so in this case.

The radiator connections and valves specified were as follows, being single pipe in each case:

Under 24 square feet	1 inch.
24 to 50 " "	1¼ "
50 to 125 " "	1½ "
125 to 200 " "	2 "
Above 200 " "	2½ "

The main riser is also intended to be used as a free-exhaust pipe, thus saving a long length of large pipe. This necessitates placing the back pressure and controlling valves of the heating system in the top floor, an inconvenience which was justified by the saving in first cost. All branches were to be attached to mains by flanged unions, and to be independently valved. Branches from risers to radiators were to have a pitch of 1 inch to the foot, and if over 18 inches long, were to be one size larger than the radiator valve.

The radiators were to be of the plain cast-iron three or four column pattern, 38 inches or 44 inches high. The use of such large radiators was due to the desire to simplify the plant and keep down its cost. It was believed that they would answer the purpose satisfactorily, as the rooms were all large and the heat would readily distribute itself. Some of the radiators on the first floor were to be covered with marble slabs, and others—marked "S" (Fig. 101)—were to have deflecting shields. Approved air valves were to be located where necessary.

The fittings were to be of the "sweep" or long-radius pattern to reduce friction. No plugs or bushings were to be used. Fittings were to be eccentric where necessary. All work 8 inches and larger was to be flanged, except couplings on low-pressure work. The valves were to be angle or gate; radiator valves, corner or angle. Valves under 2½ inches to be of best

steam metal. Pipes to be properly protected with sleeves where passing through floors, ceilings, or walls. All pipe lines were to have a pitch of $\frac{1}{4}$ inch to $\frac{1}{2}$ inch in 10 feet, in the direction of the current of steam or water. They were to be supported or anchored by approved devices at intervals not exceeding 10 feet, with due provision for expansion and contraction. All work was to be thoroughly cleaned and painted. The entire system was to be tested with 50 pounds steam pressure and made absolutely tight. Bidders on this section were to state make of radiator offered, whether three or four column, and its height. They were also to file discount sheets on material and labor, on which basis extra work could be ordered if needed.

Bidders were also to submit alternative propositions on the Paul and Webster systems of air removal. It is not intended to discuss these systems here at length. Both provide means for quickly and positively removing the air from radiating surfaces, and thus maintain it at all times at the highest efficiency. The positive means by which this is done permits the circulation of a greater amount of steam through the system, and the net result is that more work is done by a given amount of radiating surface. The system works without noticeable pressure, and therefore relieves the engines of the back pressure which ordinarily accompanies the use of exhaust steam in heating, thus permitting the use of compound engines, and making higher engine economies possible. Other incidental advantages are the removal of noises, odors, water, etc., due to air valves discharging into the rooms.

In the Paul system the air valves on the radiators, steam lines, etc., are connected to an independent system of air mains of small pipe leading to an air jet or siphon operated by steam in the engine room. This "exhauster" produces a vacuum on the air line reaching to the radiators. The air valves are so located as to collect the air to best advantage. The cooling due to the accumulation of air causes the air valve to open, when the vacuum system immediately exhausts the air from the radiator. The air valve closes as soon as it is warmed by the incoming steam. In this way the entire radiator is kept warm at all times. The Paul system does not handle any water, and is applicable to either the single or double pipe system.

The Webster system is similar, except that it handles water

as well as air, and is usually connected to the regular return system. Where the single-pipe system is used a small separate return main is run. The actuating mechanism in basement is a steam pump, which handles both water and air, and maintains the necessary vacuum.

Many other incidental advantages are claimed for both systems, and they are found quite satisfactory in service, and are coming into general use.

It was thought desirable to ask for bids in the above manner, rather than to specify one or both systems, in order that a comparison could be made between them, and a system designed to accomplish the same results with a back pressure so low as to be unobjectionable.

In determining the boiler horse-power necessary for the heating, it was assumed that each of the 24,916 square feet of radiating surface would condense about .3 pound of steam per hour as a maximum when in active service, a total of 7,475 pounds, which, being divided by 30, gives the boiler horse-power as 249.

Section G—Large Pipework and Other Apparatus.

While the exact data for this section could not be secured until the boilers, engines, and elevators had been selected, it was possible to map it out in a general way. It included the high-pressure connections from boilers to engines and pumps, exhaust connections to roof, and heating system, all the necessary covering, drains, blow-offs, receivers, controlling and relief valves, and other apparatus. The general arrangement is shown in Figs. 105 and 106.

High-pressure steam at 125 pounds was to be provided for the engines and pumps. The exhaust was to be used for heating, supplemented by live steam when required. The contractor for this section was to take the heating plant from where the contractor for Section F left it and connect it with the rest of the system complete. Each steam engine was to have a separator near its throttle of the "Sweet" or equivalent make. The engines were located so close to the boilers, and the steam pipes were so large, that the writer deviated from his usual practice of combining the separators with receivers. Returns from the separators were to be trapped into receiver. Two 300 horse-power Hoppes or equivalent feed-water heaters were to be supplied. This large

capacity was due to the fact that these heaters served also as receivers and expansion tanks. It was intended also that one could be cut out for cleaning or repairs without seriously disturbing working conditions. No separate grease extractor was provided, other than those which come with this type of heater. Three 6 by 4 by 6 outside plunger duplex pumps were specified — one for boiler feeding; one for the water service of the building; the third a reserve, capable of taking the place of either of the others. The house pump was to have a regulator to maintain a constant water level in the house tank on top floor. The contractor was also to furnish a small direct-acting air pump, of the locomotive type, for the sprinkling system. All pumps were to be properly connected, with steam, exhaust, and feed-pipe connections, blow-offs, and drains, leading to blow-off tank. The latter was to be of $\frac{3}{4}$-inch cast iron, 36 inches by 36 inches, located under boiler-room floor, provided with manhead, overflow to sewer, and 2-inch vent pipe to roof. Blow-off pipes from boilers, heaters, and miscellaneous drains, discharged into this tank. There was to be a 5 by 10 pressure reducer, guaranteed to work anywhere from a slight vacuum to 10 pounds gauge pressure, to admit live steam automatically to the heating system whenever there was a deficiency of exhaust. This regulator was to be by-passed, so that it could be taken out for examination or repairs without interfering with the working of the system. A 10-inch back-pressure valve was to be located in the eighth story on free exhaust, which would open to the atmosphere in case of excessive pressure on the heating system. A complete lubricating system was to be supplied for oiling all engines and pumps from a central source. A marble gauge board (Fig. 108) was to be erected in engine room, holding a high-pressure gauge for main steam line; a low-pressure gauge for heating main; a combination water-pressure gauge for city water main, and another for house supply system, and an 8-day clock, all to be 10-inch dial, nickel plated. The same general provisions were made as in Section F for covering, material, and workmanship. The feed-water heaters, and all high-pressure and exhaust lines in boiler and engine rooms, were to be covered with high-grade covering, equal to the K. & M. Magnesia, or Nonpariel cork, of standard thickness, canvassed and painted. Smoke flues, and domes of boilers, were to be covered with $1\frac{1}{2}$-inch blocks of same material. All high-pressure work was to be tested with 125

pounds steam, and made tight at that pressure. This contractor was to furnish all foundations necessary for his work.

Contracts and Prices.

In view of the central location and general character of the building and its tenants, and the fact that the installation was intended to be a model one, there was great competition for the work. Furthermore, as all bidders were figuring on practically identical specifications, there was no chance to claim advantages due to superiority of apparatus or methods. The figures, therefore, were strictly comparative.

On account of this competition, the work was let at exceedingly low figures, as will be seen below, the usual profits being cut out, and in some instances losses sustained. These figures should, therefore, be used with caution, and from 10 to 20 per cent. added when estimating the cost of new work. Due allowance should also be made for differences in the character of buildings and the service rendered, as determined by the peculiarities of the business carried on in it. This is wholly a question of experience and judgment.

The owners of the building under consideration did not feel warranted in supplying their tenants with a high-grade plant, claiming that the lease did not require it. As the tenants had made a ten years' lease, however, and would have to meet the fuel bills, they agreed to pay part of the extra expense.

It was not definitely decided to put in a complete plant until final bids were in, and estimates of cost of operation, repairs, and maintenance had been made. Up to that time the securing of all electrical—and even steam—service from a central station was under serious consideration. The figures, however, showed a material saving by operating their own plant, due principally to the use of exhaust for heating.

Section A—Water-Tube Boilers.—Compound engines and electric elevators having been selected, requiring 303 boiler horsepower as already shown, it was thought safe to reduce the boiler capacity to three units of 150 horse-power each, two to do the ordinary maximum service. This was equivalent to 4,955 pounds of water per hour per boiler, from feed of 212 degrees Fahr. into steam of 125 pounds gauge pressure. Water-tube boilers proving not as high in price as expected, that type was

selected, the boiler being the "O'Brien," made in St. Louis. This boiler is similar to the Heine, except that the steam and water drum is horizontal, and not parallel to the tubes; the

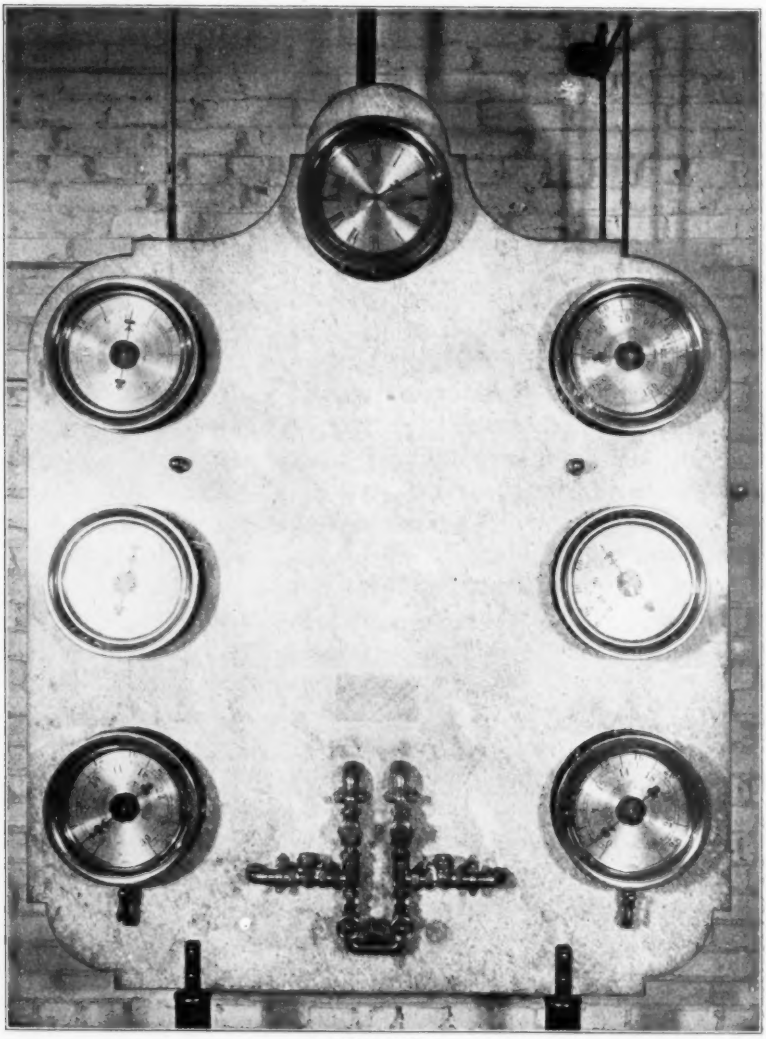


FIG. 108.

front and rear water legs are wider at top than at the bottom, and the rear leg extends some distance below the lower tubes, as a settling chamber. Each boiler contained 1,411 square feet

heating surface, including drums and tubes of the Hawley furnace, or 9.4 square feet heating surface per horse-power. The grate surface (upper grates only) was 31.5 square feet; ratio grate to heating surface, 1 to 44.78. The cost of the three boilers set up, including Hawley furnaces, brickwork, foundations, and smoke flue, as specified in Section A, was $6,288, being $13.97 per horse-power, or $1.48 per square foot of heating surface.

The cubic feet of gross capacity of building—as usually computed by architects from outside lines, and from bottom of footings to top of roof—per rated boiler horse-power, is 6,771; and 5,146 cubic feet of space heated, per boiler horse-power. The space heated included only the actual indoor space, and omitted basement.

This section, and also Sections F and G, were let to Kupferle Bros. Manufacturing Company, of St. Louis, at the gross price of $16,511.

Section Aa—Common Boilers.—The lowest bid on ordinary boilers, as per specifications (Sec. Aa), was $5,997, being $3,397 for the boilers, and $2,100 for the Hawley down-draught furnaces, and $500 for the brickwork. The four boilers aggregated 524 horse-power. The price quoted was, therefore, $11.44 per horse-power, or $0.95 per square foot heating surface.

Section B—Elevators.—The selection of the elevator system presented the greatest difficulties. The bids based strictly on the specifications seemed excessive in price, and the owners directed that the data be revised, with a view of getting more reasonable figures. The capacity of the elevators was therefore reduced. The two passenger and two southwest freight elevators had their average loads cut down to 1,500 pounds, and the four north machines to 1,800 pounds, being required to carry these loads at the speeds specified. They were also to be capable of carrying the larger loads originally specified, at reduced speeds. Wooden guide posts and strips were substituted for the steel T guides. The allowance for passenger cabs was reduced to $200, and the indicators omitted.

On this revised basis the Sprague system was selected, at the price of $22,070, including ash hoist. This low price was due partly to the conditions already named, and was made possible by arrangements made with local parties for erection, under a Sprague expert. The four north elevators were to be of the

"X 38" type, with solenoid control; and the four south elevators of the "Z" type, with pilot motor control.

To the above price should be added $3,134 for the cost of third engine, dynamo, and appurtenances, making a total of $25,204 for the elevator plant complete. The cost per cubic foot of building space (gross) for elevators alone was $.0072, and for elevators, with engine, dynamo, and appurtenances, $.0083. The cost per square foot of total floor space was: for elevators alone, $.105; with engine, dynamo, and their auxiliaries, $.12.

The selection of the electric elevator for this important installation was decided upon after a thorough investigation. They were found to be more economical in fuel than any other type—even the high duty hydraulic—on account of the light loads usually carried, and the corresponding saving in power. There was also some saving in first cost. The electrical plant afforded a much more compact arrangement in engine room. As no pumping engines were required, and no tanks or accumulators, one reserve unit answered for both the lighting and elevator plants. The electric elevator has not, as yet, fully established its reliability and low repair account as well as the hydraulic, but the latest and best types seem to leave but little to be desired in these directions.

The steam elevators were, of course, the lowest in first cost, but were not given serious consideration on account of their large fuel consumption.

Section C—Dynamos, Switchboard, etc.—General Electric dynamos were selected, the guaranteed efficiencies being:

At $\frac{1}{4}$ load	84.	per cent.
" $\frac{1}{2}$ "	89.5	"
" $\frac{3}{4}$ "	91.5	"
" full "	91.5	"

The three 75-kilowatt generators, with switchboard* and appurtenances, as specified, cost $5,135, or $22 80 per kilowatt. Speed, 260.

Gross cubic feet cared for per kilowatt of dynamo capacity for lighting	20,312
" " " " " " " all purposes	13,541

* The switchboard (Fig. 109) was built by the Emerson Electric Manufacturing Company, St. Louis.

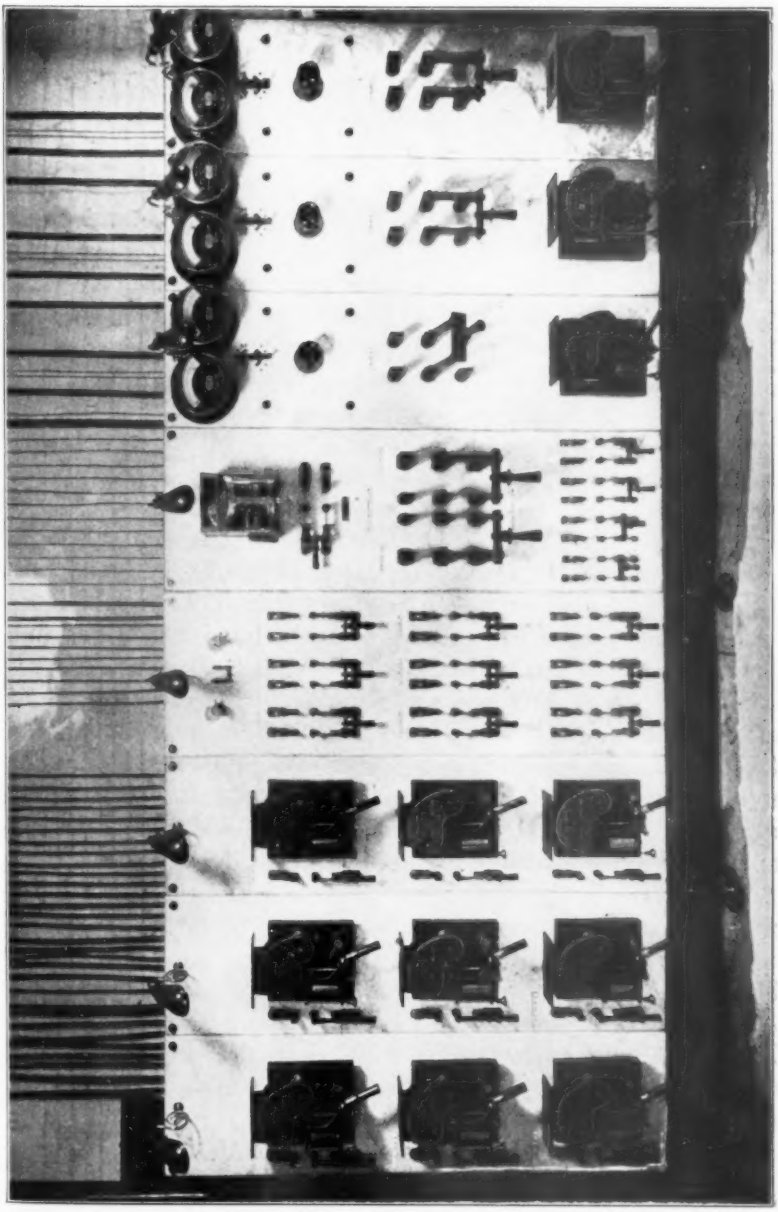


Fig. 109.

Section D—Dynamo Engines.— The three 125 horse-power engines selected were of the Imperial compound type, built by the Weston Engine Company, of Painted Post, N. Y. Dimensions of cylinders: high pressure, 12 inches diameter; low, 20 inches; stroke, 14 inches; speed, 260. Price, $4,268, or $11.38 per indicated horse-power. The same builders offered three simple engines, 15 inches by 14 inches, for the same service for $3,615, or $9.64 per indicated horse-power. This should be reduced about 5 per cent. to be strictly comparable with the cost of the compounds. Computations showed that the horse-power hours of service per year would be sufficient to make the fuel saving of the compound over the simple enough to warrant the additional investment.

The cost of engines and dynamos together was $9,403, which is $41.80 per kilowatt of dynamo capacity, and $25.07 per indicated horse-power of engine capacity.

Gross capacity of building (cubic feet) cared for per horse-power of engine, 8,125.

Section E—Wiring, Lamps, Fans, and Motors.—This work was let for $6,315, in accordance with the specifications, to the General Electric Company, their 220-volt arc lamp being selected. On the basis of 150-kilowatt capacity, the price per kilowatt was $42.10, including the entire work from the switchboard to the lights and fans, exclusive of special fixtures on first floor. Adding to this two-thirds of the contract price for Sections C and D (engines and dynamos), gives the total cost of the electric lighting plant (exclusive of elevators) as $12,584, or $83.90 per kilowatt, exclusive also of boilers and piping.

Section F—Heating System.—This work was let for $6,984, including the Paul system; satisfactory plans having been submitted for reducing the amount of radiation, size of mains, etc. The number and location of radiators were the same as specified, their sizes being reduced. Fig. 107 shows the arrangement of the Paul air and steam risers. No other changes were made, except that the first floor radiators were supplied from basement return main, instead of from an independent main. As satisfactory performance was guaranteed with standard fittings, the requirements for long-radius fittings were waived.

The following table is interesting, as comparing the bids on original specifications with the contract price for the Paul system:

	As Specified.	Paul System.	Paul System with Boilers and Piping.
Radiating surface.....square feet	24,916	18,073	18,073
Cubic feet heated per square foot radiating surface	93	128	128
" " gross " " " " "	122	169	169
Cost of heating plant.....dollars	7,689	6.984	16,511
" per square foot radiating surface..... "	.31	.39	.91
" " cubic foot heated..... "	.0033	.003	.0071
" " " " gross..... "	.0025	.00.3	.0054

It will be noticed that the radiating surface specified does not appear excessive after all, as each square foot takes care of 93 cubic feet of space heated, while each square foot under the Paul system must heat 128 cubic feet.

When a heating system is properly proportioned for hard service, with ability to heat a cold building quickly in severe weather without material back pressure, it is necessary to use ample radiating surface and piping, which, of course, mean increased first cost. In the present instance the prices indicate that the reduced surface, etc., which would do the same work under the Paul system, saved enough money to not only pay for all special apparatus, royalties, etc., but to make a considerable saving to the owners besides.

Section G—Large Pipework and Apparatus.—This was let for $3,239, being $7.20 per unit of boiler horse-power, and 18 cents per square foot of radiation contracted for.

Total Costs.—The total cost of the mechanical plant ($54,299) may be divided as follows:

Per cubic foot of gross space.....	$ .0178
Per cubic foot of space heated.....	.0234
Per square foot radiating surface contracted for.....	3.00
Per rated horse-power of boiler capacity.....	120.66
Per indicated horse-power of engine capacity.....	144.80
Per kilowatt of generator capacity.....	241.32

The total cost of building, including mechanical equipment, ready for the tenant's use, was $301,000, or 9.9 cents per gross cubic foot, and 13 cents per cubic foot heated. The cost named included $20,000 for the use of the party walls of the adjoining buildings, and $10,000 for Luxfer prism glass, and $9,000 for a sprinkling system. The value of the ground was $200,000, making the total investment a trifle over $500,000.

Tests and Performance.

During construction and erection the work was under the supervision of the engineers, and on completion it was submitted to careful inspection and tests to determine whether the contract guarantees had been met. Since going into service the plant has been under the observation of the engineers, so that close track has been kept of its performance.

Section A—Boilers.—These were submitted to a number of evaporative trials to determine their capacity and efficiency. The details are not given here, as the methods and results are of no special interest. The boilers, however, give good service, particularly in capacity, dryness of steam, and smokelessness.

Section B—Elevators.—An exhaustive detailed test of capacity, speed, and efficiency was made on No. 5 elevator, type "X 38," situated in the northeast shaft, and on No. 1, type "Z," located in the southwest shaft. Simultaneous readings at 2½-second intervals were taken of the voltage, speed, and current—the latter being checked by two instruments, two observers reading independently. The live loads upon the platforms were obtained by weighing.

The speed of travel was calculated by dividing the revolutions of the armature by the known ratio between speed of armature and speed of car. The power consumed per trip was obtained by plotting the curves of current readings (Fig. 110), the area of which—multiplied by the voltage—gives the watt hours per trip. This was reduced to mile of travel by multiplying the watt hours per trip by the fraction that one round trip is of one mile. The following table gives the results:

Elevator.	Live Load, Pounds.	Speed Up, Ft. per Min.	Speed Down, Ft. per Min.	Kw. Hours per Car Mile.	Remarks.
Z, No. 1	147	271	268	3.14	
"	908	256	256	3.14	
"	2,997	153	157	4.22	Slow speed.
"	2,997	256	290	4.19	High speed.
"	4,000	147	192	4.05	Slow speed.
X 38, No. 5	147	153	153	3.14	
"	1,008	157	155	3.19	
"	4,000	152	160	3.34	
"	5,012	153	160	3.81	

The observed data from the X 38 machine are plotted in Fig. 110. It is interesting to note the current of return to the line with light loads ascending and heavy loads descending. It will be

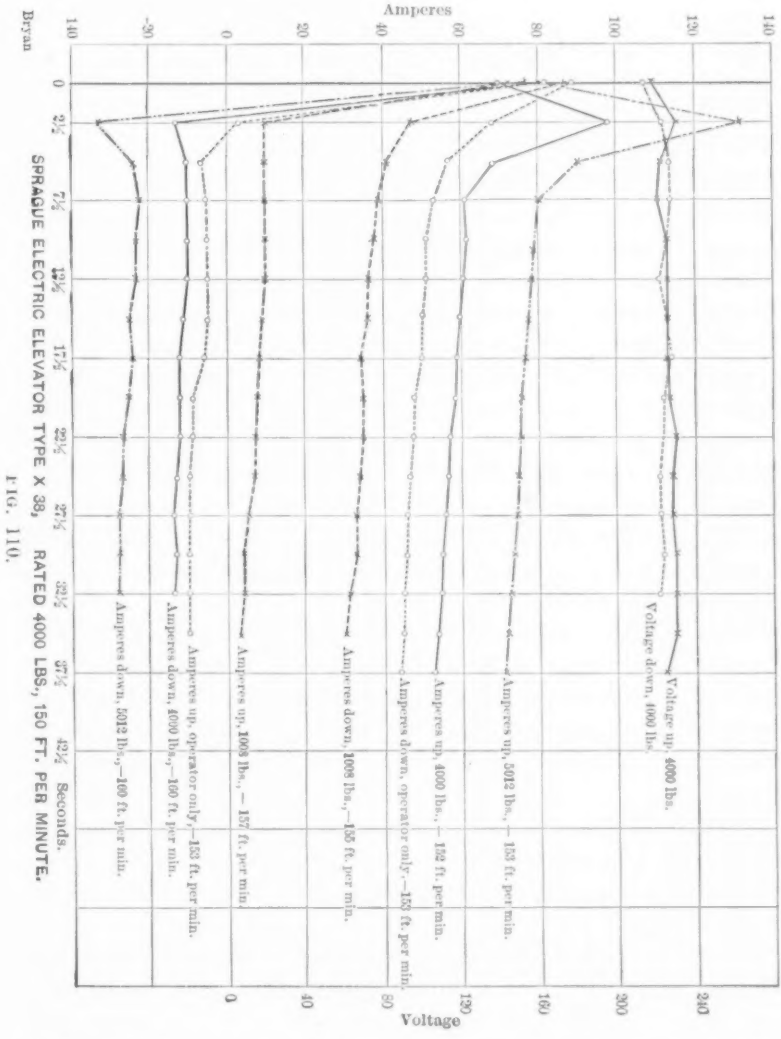


FIG. 110.

noted that No. 1 elevator has a down speed, with operator only, of 268 feet per minute. With the same counterbalancing, it carries a load of 3,000 pounds up at 153 feet per minute, with motor operating on slow-speed notch of operating lever. When operated at high speed, this elevator lifted 3,000 pounds at a

speed of 256 feet per minute, and with an economy per round trip slightly better than when operated at slow speed. This elevator carried 4,000 pounds up at 147 feet per minute, when operated on the slow-speed point of operating lever.

The No. 5 elevator has a down speed of 153 feet per minute when running with operator only, and carries its full rated load of 4,000 pounds up at a speed of 152 feet per minute. This elevator carries 5,000 pounds up at a speed of 153 feet per minute.

It is particularly interesting to note that the efficiency in kilowatt hours per car mile increased but slowly with increased loads. The kilowatt hours per car mile of travel are equal to good average practice, and will probably improve after longer operation of the plant.

The adjustment of the starting devices on the elevators was found to be for a 2½ to 3 seconds' start. This made the starting current exceed the running current by more than the 50 per cent. specified, which was based on a 5 seconds' start. When the starting devices were readjusted for a 5-second start, this condition of the specifications was practically met. It is a question of judgment, therefore, whether the tenants prefer a slow and easy start and a smooth-running electric plant, or a quick start and a correspondingly severe and sudden demand upon the dynamo and engine.

Two dynamos were operated in multiple, running both elevators and lights. With a variation of load from 250 to 600 amperes, the ordinary variation of voltage was from 220 to 226, and the maximum, 218 to 228. With a variation of load from 200 to 500 amperes, the average variation of voltage was from 216 to 220, and the maximum 214 to 222. This variation did not affect the lights sufficiently to be noticeable, unless the observer's attention was called to it, but has been further improved by a more careful adjustment of the elevator-starting devices.

The wattmeter on switchboard shows the electric power used by the eight elevators from July 12 to 27, 1898, inclusive, or 138½ hours' operation, to have been 1,920 kilowatt hours, or an average electrical horse-power per hour of 18.6 for the plant, or 2.3 horse-power per elevator.

During the month July 12–August 12, 1898, the hours of operation were 283.5, and the average electrical horse-power was 19.2, or 2.4 per elevator. Unfortunately, no records were

kept of the miles of travel during this time, but they may be roughly approximated as follows, taking the first period mentioned:

Assuming an average efficiency of 4 kilowatts per car mile, the 1,920 kilowatt hours mean a travel of 480 miles. The travel of the eight elevators per round trip is 1,608 feet, or .304 mile. Dividing 480 by .304, the number of round trips is found to be 1,579 in 138.5 hours, or 11.4 per hour, or $5\frac{1}{4}$ minutes per round trip. As the average travel is about 201 feet, and the average speed about 200 feet per minute, the round trip should have been made in about a minute. This indicates that the elevators averaged about $\frac{1}{5}$ the time in motion, instead of $\frac{1}{3}$ to $\frac{1}{2}$, as originally assumed for maximum service.

It is gratifying to note that the power actually required to operate the elevator plant is considerably less than originally estimated. This is due to the use of apparatus of higher efficiency than assumed, to smaller percentage of time in motion, and to the loads contracted for being less than specified.

The data in the following table, showing the relative performance of three elevator plants in St. Louis, will be found interesting.

A and *B* are high-pressure hydraulic plants of the most modern type, plant *A* having a high duty compound crank and fly-wheel pumping engine, and *B* an ordinary direct-acting compound pump. The Hagardine-McKittrick plant is shown in column *C*, and is furnishing arc and incandescent lights from the same bus bars. The fuel and water results are computed from the guaranteed efficiencies of boilers and engines; the elevator data, however, being based on actual test.

	A.	*B.*	*C.*
Cost of coal per ton of 2,000 lbs	$1.35	$1.50	$1.40
Water evaporated per pound coal	8.	8.	7.64
Water required per I.H.P. hour	.....	54.21	25.
Coal " " " " " "	.....	6.78	3.27
Kw. hours per car mile—average load	.....	.....	3.165
H.P. " " " " " " " electrical	.....	.....	4.24
" " " " " " " indicated	.....	.....	5.30
Lbs. coal " " " " " "	24.	58.54	17.33
Cost of same—cents	1.62	4.39	1.21
Average I.H.P per elevator July 12–Aug. 12, 1898	.....	.....	3.00
Cost of coal per elevator per hour—cents	.....	2.92	.69
" " " " " day of 10 hours—average cents	.....	29.24	6.87
Cost on basis of a round trip every 9 minutes—cents	4.13	9.75	2.75

The unfavorable showing of plant *B* is due almost wholly to the low duty pumping engine. The ordinary hydraulic plant, using low-water pressures, and ordinary duplex pumps—sometimes not even compounded—is still more wasteful in fuel. The favorable showing of the electric elevators is due to good counterbalancing, reduced power with loads under the maximum, and compound steam engines. The tests of kilowatt hours per car mile were made with stops and starts at terminals only. The power required will, of course, be larger when many intermediate stops are made.

Section C—Generators.—Exhaustive tests were conducted on the dynamos, covering capacity, heating, efficiency, and general operation, with varying loads, both under and above rating. Very satisfactory performance was found, and complete conformance with the rigid requirements of the specification. The efficiency of the generators at various loads was computed by means of indicator diagrams and switchboard readings, using carefully calibrated instruments.

Section D—Engines.—These were tested for capacity, regulation, and general performance, and were found acceptable. The following results were secured:

Indicated horse-power	126.74
Electrical horse-power at switchboard	102.37
Efficiency of engine and dynamo unit—per cent	80.77
Water rate per indicated horse-power hour—pounds	26.79

The latter was a trifle higher than the guarantee of 25, but the test load was a little too large, and the engine was a trifle out of adjustment. The engine efficiency was determined by a four hours' trial, measuring the water which entered the boiler, and collecting the condensation in pipe system by means of separators, drains, and traps, and deducting its weight from that of the water which had been pumped through the measuring tanks.

Section E—Wiring, etc.—This section required but few tests, those made covering the points of insulation, resistance, and the operation of the 220-volt enclosed arc lamps, and the accompanying mechanism, switchboard, etc., all of which was found satisfactory.

Section F—Heating System.—Tests on this part of the plant have not yet been made, as they could not be carried on success-

fully until cold weather. The system, however, was in use during a number of cold days in October, and its performance was satisfactory. The Paul exhausters maintained a vacuum on the exhaust line of 15 to 20 inches, and the heating was done with ½-pound back pressure, and, no doubt, could have been done with less. The circulation throughout the system was found to be complete, and all radiators hot, within twenty minutes after turning steam of ½-pound pressure into the cold system.

Section G—Large Pipework.—No special tests were required on this work, other than the observation of the actual performance of the different units in service. All were found to do their work satisfactorily.

In conclusion, it is interesting to note the actual working efficiency of the plant as compared with that of the building formerly occupied by the same tenants. There the lighting, heating, and elevator service were less than half that of the new building, as were also the floor space, cubic feet, and actual business transacted. The old building employed old-style boilers and furnaces, simple dynamo engines, and steam elevators. It was a corner building, however, and required but little artificial light during the day. In spite of this, however, the improved apparatus in the new plant has kept the coal consumption down to only about 25 per cent. more than in the old building, although the work done has more than doubled.

APPENDIX.

Exhibit A.

Recapitulation of Data as to Costs, Ratios, Performance, Etc.

In using these figures, or comparing them with others, reference should be made to the explanations in the body of the paper, to determine the exact meanings.

Boiler Plant.

Square feet heating surface per rated horse-power	9.4
Ratio grate to heating surface	1 to 44.78
Cubic feet of building—gross—per rated horse-power	6771
" " " " heated " " " "	5146
Cost of complete boiler plant, per horse-power	$13.97
" " " " " " square foot heating surface	1.48
Cost of ordinary boilers to do the same work, per rated H.-P.	11.44
" " " " " " " " " " sq. ft. H. S.	.95

Electric Elevator Plant.

Cost per gross cubic foot of building—elevators alone	$.0072
" per gross cubic foot of building—elevators with engine and dynamo	.0083
" per square foot total floor space—elevators alone	.105
" per square foot total floor space—elevators with engine and dynamo	.120
Kilowatt hours required per car mile of travel	3.14 to 4.22
Average electrical horse-power required per elevator	2.35

Dynamos and Switchboard.

Cost per kilowatt of rated capacity	$22.80
Gross cubic feet cared for per kilowatt, for lighting	20,312
" " " " " " " " all purposes	13,541

Steam Engines for Dynamos.

Cost of compound engines with foundations, per rated H.-P.	$11.38
" per rated kilowatt of dynamo capacity	18.97
" of simple engines to do the same work, per rated H.-P.	9.64
Gross cubic feet cared for per rated horse-power of engines	8,125
Water rate—pounds per indicated horse-power per hour	26.79
Cost of dynamos and engines per rated kilowatt of dynamos	$41.80
" " " " " " " horse-power of engines	25.07
Efficiency of dynamo and engine unit, per cent	80.77

Wiring, Lamps, Fans, and Motors:

Cost per kilowatt of dynamo, lighting capacity	$42.10
" including lighting, dynamos, and engines	83.90

Heating System.

Cubic feet gross space per square foot radiating surface	169.
" " heated " " " " " "	128.
Cost per square foot radiating surface	$.39
" with boilers and piping	.91
" per cubic foot gross space	.003
" with boilers and piping	.0071
" per cubic foot heated space	.0023
" with boilers and piping	.0054

Large Pipework and Apparatus.

Cost per rated boiler horse-power	$7.20
" " square foot of radiating surface	.18

Complete Mechanical Plant.

Cost per cubic foot of gross space	$.0178
" " cubic foot of space heated	.0234
" " square foot radiating surface contracted for	3.00

Cost per rated horse-power of boiler capacity	$120.66
" " indicated horse-power of engine capacity	144.80
" " kilowatt of generator capacity	241.32

Completed Building.

Cubic feet of space, gross	3,046,727.
" " " " heated	2,315,612.
Square feet of floor space	210,116.
Cost of building per cubic foot gross	$.099
" " " " " " heated	.130

DISCUSSION.

Mr. A. K. Mansfield.—The portion of this paper relating to the economy of the engines is of particular interest to me, for I am engaged in the production of engines, which are used in similar mechanical plants. Moreover, we have lately made a careful test of an engine—at the works of the Buckeye Engine Company—which is so nearly of the same size and power, and test was made under so nearly like conditions, that it may interest our members to compare the result with that given by Mr. Bryan.

Our engine had cylinders 11 inches and 19 inches in diameter, its stroke was 16 inches, and revolutions per minute 225. It was run non-condensing, with initial steam pressure of 125 pounds above atmosphere, and when delivering 180 horse-power by card, its rate per horse-power per hour was 20.4 pounds.

The exhaust was caught by a surface condenser and weighed accurately.

The tests were made with the coöperation and assistance of two government engineers, Messrs. J. E. Woodwell of Washington and H. Adams of Baltimore, the engine being one of several which we have recently built for the United States Government.

It seems fitting that this result should go on record here, in order to make it clear that the economy of operation of a mechanical plant of a commercial building may be much better than that quoted by Mr. Bryan.

Mr. A. F. Nagle.—I desire to express my gratitude for the completeness with which Mr. Bryan has presented the data of his paper. It is of great value, and more of a similar character pertaining to other plants would be very desirable.

I wish, however, to refer on page 271 of the paper, where the

author has required the engines to regulate under a steam fluctuation between 100 and 125 pounds pressure.

I wish to ask the author why he makes provision for such fluctuation of pressure? We are all aware that there are steam boiler damper regulators which maintain a pressure within one or two pounds, and in such a complete plant as Mr. Bryan has designed for this particular work, why does he not avail himself of a modern damper regulator to maintain uniform pressure? It is my opinion that not only has the damper a value in giving us all the advantages of uniformity of pressure, but that it is also conducive to economy, fully paying for the cost thereof.

Anticipating, perhaps, Mr. Bryan's answer to this, that the Hawley Down-Draught people do not like to have an automatic damper attached to their furnaces, I will relate my experience with the Hawley people on this subject. In a plant which I am now installing with the Hawley furnaces, they advised me not to put in the automatic damper. The reason they give is that the closing of the damper throws the hot gases into the fire room. Discussing this matter with them, they agreed that they would try and attach the damper regulator to the furnace doors; but in a letter received from them only a few days ago, they advised me that this is impossible, but that if I will make the damper fit *very freely* in the flue, or adjust it so that it will not close up tight, the same end will be attained as if attached to the front doors. While I am not quite satisfied with this explanation, it seems to be the status of the state of the art at the present time, namely, that the damper regulator must remain in the chimney flue instead of being attached to the front doors.

I wish to reserve for a little later communication some remarks I would like to make on the enclosed arc lamp selected by Mr. Bryan for this particular plant.

Mr. Jesse M. Smith.—While I think the author of this paper has gone into unusual and unnecessary detail for a paper to be presented to a society of this kind, still it strikes me that in other particulars he has not given full information. For instance, on electric elevators, he simply refers to them as "No. 5, type X, 38," and "No. 5, type Z;" that may communicate some information to some people, but not very definite information to the majority of people, I think.

Mr. Reginald Pelham Bolton.—I have some remarks to make upon this subject, which is one that I have made to some extent

my own in this city; and if I might depart from the paper first to answer the question of Mr. Smith, who has just spoken, I think I could give him some light upon these types of elevator machines. They are both drum pattern machines, worm gear, driving direct, and the type Z machine, if I am not mistaken, is a double-drum machine; that is, with two sets of worm gear. In this paper there are, as Mr. Smith has remarked, a number of things that seem to have been omitted, while others have been considered with very great detail. The questions that the author put down for primary consideration included some which I should have thought would be disposed of without any consideration at all—as to whether power should be distributed by belting, shafting, rope-drive, or compressed air, in a building of that character. The consideration of the other points follows in very good order, although in this city the primary consideration is always the heating system. More especially in the very tall buildings down town the heating is the primary necessity; that is an absolute necessity to the life of the tenants, and it is usually, in the bigger buildings we have here, the largest item. Once that is considered, the boiler power is fixed, and everything else should come within the limits of the steam that is used for heating purposes, and in a building of this particular size, it might be a very nice question as to whether they would not be very nearly balanced. But the author seems to have arrived at the conclusion that his services would exceed the amount of steam required for heating the building, and to have proceeded on that assumption. That leads directly to the question of the amount of heating surface that he provided for this building. I notice that his original specifications called—and I consider very properly—for an amount of 24,000 to 25,000 square feet of radiating surface. That was arrived at by a series of calculations which are made on the ordinary system,-the data for which he gives us here; and by looking into the plans of the building without its elevation, it would appear to be not in excess of the requirements. Even that, however, was arrived at with an assumption that the outside temperature of the atmosphere did not fall below 10 degrees below zero; whereas I assume in St. Louis, as here, you are very liable to get zero temperature with a very strong wind, which is the most serious condition. Now, I have found in my practice in this city that the wind is a far more serious element than the actual temperature itself, and that an allowance has to be made

for it according to the conditions of the height of the building. That element I do not see taken into account here. I might say, in connection with that, as illustrating the difficulties that are met with on account of the wind, that in one of the tall buildings that I have in charge, the first story and the second story were considered to be sufficiently protected by the surrounding buildings, and the heating surface was proportioned accordingly. It happened that at the back of the building there was a V-shaped court (formed by this tall building, which is 208 feet high, and also a building at the far end of it), opening to the southeast. At certain points of the wind, not only direct from the southeast, but also from the northwest, the wind would back up in the bottom of that court and force its way into the windows of the first and second stories through leakages there, at a most undue rate of speed. Through an anemometer four miles an hour speed was registered just inside the window crack, and that makes a condition of service which renders your heating surface a very doubtful element. It was a curious fact that heating surface was taken temporarily from the fourteenth story of the building and transferred down to these first and second stories, and the effect on the fourteenth story was not very deleterious.

The author appears to have allowed the bidders on his steam-heating plant the option of taking his heating surface, or of changing it to a less amount on a system of removing the air by the Paul system, provided they brought up a guarantee that they would heat the building. That put a great deal of power in the steam-fitter's hands, and the result is that he gets in his building now the advantages of the Paul system, which are undeniable, but he only gets 18,000 square feet of heating surface. Now, that reduction of the heating surface was evidently arrived at, according to my view, on the steam-fitter's part, by taking chances which the author himself was not prepared to take. For instance, I think you could find that the reduction from 25,000 to 18,000 feet was arrived at in this manner; that the exposed side of the building, that the author brought into his computation, by the + 10 W in Carpenter's formula, was neglected altogether. There are 750 square feet allowed in connection with the exposed side of the building. Now, the exposed side of the building was not really exposed, because there was another building alongside. The author assumed that the building was unoccupied, and he had assumed for that wall as low a temperature as 32 degrees.

The riser pipes, also, were taken into account, and about 500 square feet were knocked off the radiators on account of the riser pipes. Now, the author, I think, took a heat-unit distribution through his radiating surface, of 288 units per square foot per hour. I think that the probability is that the steam-fitter took 300, which would add about six per cent. to that value of his heating surface, and would knock off another 1,200 square feet. And, then, the common computations and assumptions of all heating men are based generally upon zero outside and 70 degrees inside, whereas the specification issued by the author permitted a more generous assumption. That made considerable difference, and I should expect an ordinary steam-fitter's computation would knock off as much as 3,000 or 4,000 feet on that account. Those items account for the reduction of heating surface. It is possible that 18,000 square feet would heat that building fairly well, but their relation, standing as they do at 1 square foot to 100 cubic feet of contents, is in my judgment altogether too small. We would not permit that in this city. Of course the computations and comparisons based on cubic feet of contents to square feet of surface are mainly unreliable, and no definite results can be expected from them. Basing steam-heating surface on the contents only, you will find such cases as I found about a year ago—a small room on the eleventh story which had four windows exposed to the southwest, and the steam-fitter had placed in it the ordinary amount of surface on basis of 1 foot to 70 cubic feet. It was utterly insufficient. A rough computation made on the spot on the glass surface, showed him that his proportion should be 1 to 35, and that amount proved to be necessary.

The saving clause on the system that has been put in this building is the use of this Paul device for reducing the back pressure on the machinery, and circulating at a very low tension. That is a very meritorious system indeed, and it seems to my mind practically the only successful way of dealing with a condensing system, such as the heating system now is—that and the Webster system, with which I have had very great success. I might say it has always appeared to me that the problems connected with the plant of a commercial building are precisely on similar lines to those of a vessel. You may regard a big building as being a ship at sea. It stands isolated from its neighbors. You bring into it coal and water supply, and you have certain services which must be kept running a certain number of hours a day,

and they are as necessary to the life of the building as the running of the engines are on board a ship. The condensing system, which is the heating system, running as it does for about one-third part of the year, is also an absolute necessity, and should be taken into account. But if the condensing system for a third of the year imposes on the rest of the machinery a condition of back pressure which puts much of it at a disadvantage, then a building on these lines cannot be regarded as an economical or well-arranged building. The common practice, even in our big buildings down town, has been to put in ordinary gravity systems, or a single-pipe system, and the pressures with these to-day down town range as high as ten to fifteen pounds, and that is exerted as a back pressure on compound pumps and compound engines in some cases, and the curious result follows that the steam consumption becomes cumulative. The pumps, as the day's work increases, use more and more steam, until finally the condition is reached at which they exhaust steam to the atmosphere, wasting steam all the time, while they should normally be only supplying enough to heat the building—throwing away steam all the time that it is required in the building. In other cases they have had to cut the pumps adrift and let them exhaust direct to the atmosphere and not into the heating system at all, in order to do away with the back pressure. Therefore, the adoption by the author of a system of reducing the back pressure was a very meritorious feature of the building.

I would like to remark on the preference that is given to the water-tube boilers. I think that the author did not give sufficient prominence here to the feature that where you have to place boilers right under a building, with so many human lives over them, the risk of destruction of the building by explosion is a primary consideration, and considering the relative cost of boilers to the other plant, the extra cost involved in water-tube boilers is a flea-bite compared to the security that they afford. While the water-tube boiler is liable to accident, as well as any other, the idea that the accident to which it is liable is only of a nature that kills the engineer that operates it, and not the tenants in the building, is one that sometimes appeals to all of us.

I might make a remark on rather a curious feature that the author brings forward here as to the tenants paying for a portion of the plant. That seems a very novel proposition to us in this city. Here the owner cannot put in enough machinery to oblige

his tenants. Sometimes when his tenants get in he has to add to his machinery to oblige them.

In specifying the requirements of the boilers, we come to a point that I have had some considerable success in; that is, in drawing the specifications so that the boilers shall all be presented on a reasonably even basis. It does not do to lay down heating surface, for then you get all the controversy brought up as to the relative value of heating surface. It has appeared to me to be better to draw a specification in which I first lay down chimney conditions; secondly, the coal that I was prepared to supply for the work; and thirdly, the amount of coal that I desired to burn on the grate per square foot of surface, and then call for the required amount of evaporation, just as the author does here, from a certain temperature, which in our practice here is 208 degrees, to steam of a certain pressure. The result has been, in my practice, that bids for the boilers have come in on a remarkably even scale. In a recent case where I drew specifications for 500 horsepower, the bids all ranged within a very small percentage of each other, and the boilers got a fair representation, each manufacturer, of course, basing the value of his boiler on what he estimated he could do under those given circumstances. But it does appear to me that without the chimney conditions the boiler bidders are at a considerable disadvantage.

Passing on to the next feature that is considered, the elevator system, the author rightly says it is the most complicated problem he ever encountered in this matter, and I think that it is very apt to be complicated, on account of the extreme division there is between the different interested parties on the subject. I happen to have waded through the thick of this question, and to have ascertained a good many facts upon it which I hope at a later period to present to the Society, but this paper brings up some of them which I might present at this moment as bearing on this particular installation. The elevators which were installed in this building were those with a very heavy load and a very moderate speed, and they were of this screw type, with drums. They do not altogether compare with our elevators for passenger service in this city, where we are running on a regular schedule, and running at a higher rate of speed, but still the same general considerations apply. I notice that the author, in specifying his hydraulic elevators, called apparently for a type of hydraulic elevator which would appear to us here to be disadvantageous; that

is to say, he said that the water pressure for the hydraulic was not to exceed 750 pounds at the pumps, evidently a high-pressure system which has been adopted in one case in St. Louis, and only in one case, with moderate success, in this city, the American Tract Society's building, and he specifies an accumulator instead of a pressure tank which, of course, would be necessary with that very high pressure. That made the invitation for a hydraulic system somewhat abnormal, and I should imagine considerably to the disadvantage of the hydraulics. The results that the author presents in the table on page 291 are subject to some criticism. Table *A* appears to relate to a high-pressure hydraulic plant of special type, which is also referred to in an earlier stage of the paper, and which I take to be the Cupples plant in St. Louis, if I mistake not. I do not wish to do any injustice to the Cupples plant, but it is a plant consisting of forty elevators in a very extensive series of warehouses, carrying very heavy loads at very slow speeds, with direct-acting rams, and at a pressure of 745 to 750 pounds per square inch. The gear of these elevators is only 2 to 1, and consequently the internal friction of the machine is very low; indeed, also the operation of forty elevators produces a fairly even load upon the generating plant, and can in no sense be compared with any other hydraulic or electric plant in such a building as is under discussion. I had the figures of the Cupples plant, and made an analysis of it, which is of too extensive a character to enter into to-day, but those are the prime features of it. The average lifting speed is only 150 feet a minute, I believe.

The second plant, "B," appears to be an ordinary direct-acting compound pumping system, and the author gives as the result of the plant—I do not know with what degree of closeness it has been ascertained—the figures in column "B." I have some figures, also, bearing on that—comparative figures—of which I will speak in a moment. And then in "C" he gives us a column of the results ascertained from the use of the electric elevators in this building, noting at the same time that the fuel and water results are computed from the guaranteed efficiencies of the boilers and engines, the elevator data being based on actual test. That is very good as far as it goes, but a good deal of this column of information is attributable to the very questions that are computed only from guaranteed efficiencies, and therefore are not definite. I notice at an earlier stage of the paper he mentions

that the engines did not reach their guaranteed efficiency. Now, that brings this question up: In taking the guaranteed efficiency in a steam engine, which is driving an elevator load of fluctuating character, you will generally find a considerable amount of disappointment. I have on actual test made a running trial of a simple engine driving five electric elevators on a schedule, on a very light average load in the cars. That engine was installed under a guarantee of 33 pounds of steam per indicated horse-power per hour. It was an engine that was working under the Webster system, in which there was no back pressure in the house heating. It appeared as the result of the test that the average steam consumption of that engine was 56 pounds per indicated horse-power per hour. That was entirely due to the fluctuating character of the load, which load varied all the way from zero to 25 per cent. over the maximum power of the generator, at intervals of one and two seconds, so that the steam consumption cannot be assumed and the coal consumption cannot be assumed from any guaranteed efficiency on an engine working such a load as that. It would be equally unreliable to take the guaranteed efficiency, or guaranteed work of a hydraulic pump, say a compound pump, for which the makers will guarantee 66 pounds per pump horse-power per hour, for if you go to test that on an elevator system you will find that the pump is creeping at times, and overrunning at times, and your steam consumption will creep right up on you, and you will be very much disappointed in your guarantee. It is simply a question of steady and irregular loads, and working conditions also. Now, the pounds of coal given here I take it are related to the car mile and are extraordinarily low under this computation —17.73. As an actual result of the tests that I have, I got all the way from 40 down to 30 pounds of coal per mile run with the elevators, and that result, compared with the tests that have been made on a compound pumping plant, puts the electric elevator plant on parallel lines with the compound hydraulic plant. It might perhaps be of interest if I gave the result of a comparison that I have made in this city, based on a service of eleven miles an hour. This, of course, is passenger elevator work—not carrying freight at all. It is at a speed of about 415 feet a minute. The compound pump, using 70 pounds of steam per pump horse-power per hour, gives a cost per mile of 9.44 cents, and I have found that an electric elevator, when fairly loaded with 800 pounds

average load, comes out about that same figure. I might incidentally mention that as far as hydraulics are concerned, the very successful system of pumping by double pressures at the Bowling Green building has shown itself to be the most economical way of handling the elevator problem. That system has to pump with a triple expansion pump all the water required for the elevator service under a sufficient pressure to raise the average load in the car. In that case 120 pounds is the pressure, and the average load elevated by that 120 pounds is 800 pounds in any car, and at that load it will run at its full speed. Now, if the average load is exceeded, the operator throws the lever over into the further notch and he takes in high-pressure water. That high-pressure water is provided by a compound pump drawing its suction from the 120-pound tank and pumping into a 210-pound tank, and the 210-pound water is piped over on the same valve, the valve being provided with a little additional movement by which, when the lever goes over to the forward notch, it passes over a second port and admits high-pressure water. I made a test of that method of pumping, which has not been previously made known, and which gave some really very remarkable results, under circumstances that would compare in a very even fashion with this particular plant under discussion. I had ordinary high-pressure duplex pumps, supplied with 85 pounds of steam, and delivering over into a condenser supplied with a liberal amount of city water, producing complete condensation, and the condensation was weighed into tanks, so that the test was of a very precise nature. The elevator was loaded with cast-iron weights, weighed in and out, in order to compare the results. The conditions were such as I have named; that is to say, the two pressures were what I have already mentioned, and the average load was 780 pounds. We started out with an average load in the car, taking all the water from the lower pressure. We got a consumption of 192 pounds net of steam to the mile. That is obtained by an ordinary elevator plant with a non-compound pump. By successive increments of load, which required the use of high-pressure water, that was brought up to the final point of 387 pounds net to the mile, by the gradual increase of the use of the second, or high pressure, pump, and the remarkable fact was that the high-pressure water seemed to proportion itself to the amount of load in the car, by a sort of injector action, the water going in and uniting with a portion of low-pressure water, so that the water used

from the high-pressure tank is directly proportional to the increase of load, from the average load upwards.

I apologize for trenching so much on the time of the Society, but the subject is a very interesting one.

I would like to add further that in the selection of the engines and generators for this work, while I have not any criticism to make on the subject of putting in a large machine, it was not necessary, outside of the use of electric elevators, to put in machines of this size. The lighting load of this building could have been carried with two generators of a smaller size than the two finally selected. Therefore, the cost of the elevator plant should be charged not only with the third unit, but with the increase of the two first ones. That is a matter that is constantly coming up in my practice, and I think that the best way with these things is to treat them entirely fairly. The electric elevator is a good thing, but like many other good things they come high and you have to pay for them. I have no objection to paying for them if the advantages are relative to the cost.

The author speaks of the question of the cost of repairs. It is an unfortunate thing that the cost of repairs of electric elevators is high. The nature of the apparatus is such that the wear and tear upon them are severe, and I found that that is a very serious item to be added to the cost of running the elevator plant. The selection, by the author, of a compound engine I cannot altogether follow. Even with the absence of back pressure, the variability of the load on a compound machine, in my judgment, has too great an effect on the operation of the engine to justify the extra cost of a compound, and we get more reliable results out of a single engine, especially when you work the building, as we often do, short-handed.

I might add to the corrections of the author, that on page 294 the figures are probably inverted there by error—cost per cubic foot of gross space and cost per cubic foot of heating space are inverted. I think the error would be worth clearing up there.

Mr. George Hill.—The paper, it seems to me, is similar to the one that was criticized yesterday in that it is like a meal prepared without salt. There is a good deal of information in it, but the operating economies are not stated. To remedy that defect I telegraphed Mr. Bryan yesterday, and have a telegram from him in answer, which I will refer to in the proper place.

The clause on page 243, in which the author separates engineer-

ing work into plumbing, sanitary and structural arrangements, and the mechanical system and electric plant, seems to me to give an undesirable subdivision. Some six years ago I published the opinion that all of the engineering work of the building ought to be the design of one engineer, in order to produce a harmonious result, and all the experience that I have since gained only serves to emphasize that view.

On page 245, in mentioning the various possible combinations of heating and ventilating systems, the author says: "If hot air or indirect, shall it be by the natural or hot blast system?" I have had some experience with a combination office and lodge-room building of about the same cube as this building, and while the heating was entirely successful from the heating point of view, yet from the tenants' point of view the indirect system of heating is not a success, nor should I judge it to be desirable from the owner's point of view, in that the amount of space required to accommodate the ducts is very large.

Further down the page the author says: "It is impossible, therefore, to draught general rules." That seems to me a little at variance with the facts; we must be governed by general rules, varying from them as special conditions dictate.

On page 246 the author says: "In the designing of this plant a prime consideration, and one which always presents itself, was the necessity of getting a high grade installation for as little first cost as possible." It would seem that as all buildings are judged finally from the owner's or economical point of view, to produce revenue, the best plant is obtained when the operating cost, charging into that the interest and depreciation on the plant, is at a minimum.

On page 250 the author mentions the fact that of course there are some features that could not be definitely decided upon in advance. That is entirely at variance with my practice. We decide in advance what is needed and then obtain from contractors their tender to do this work, and it indicates a lack of experience where an engineer demands of the contractors four or five different sets of bids in order to enable him to make up his mind as to what is the best plant to put in the building. This is further evidenced on page 262 and on page 283, where there are requirements stated for the elevators as indicated by the tenants, and then practically cut in two when they found out what those requirements meant in dollars and cents, showing that either the

tenants did not know what they were talking about, which is probably the case, or else the engineer was willing to let them ask for absurd requirements so as to get the credit of reducing later on.

Referring to page 250 and part of page 254, the notice to bidders, general clauses relating to the specifications, etc., I find nothing therein mentioned as being different or new or in addition to what has been my practice for ten years past, and what has been good practice in specifying for similar works, I think for probably fifteen years past, and it would seem that these parts might very well have been omitted.

The computations in regard to the boiler horse-power, etc., pages 255 and 256, are very interesting mathematically, but a more certain method of ascertaining the requirements, etc., of the plant would have been to have compared it with a somewhat similar plant designed by the author, making the necessary allowances for special conditions.

The lay-out of the boiler room, shown on page 258, Fig. 105, seems to be open to criticism on the score of excessive length of steam pipe. My practice for such a lay-out as this would have been to have run the main steam pipe through just to the right of the branch to the middle engine, and put in, instead of an eight-inch pipe, a nine-inch pipe, so as to have a section of pipe acting as a receiver. This would have eliminated bends, decreased friction and decreased cost.

On the following page, Section G, the arrangement of piping and feed-water heaters on the exhaust main is different from my practice, introducing a considerable number of unnecessary bends and an excessive length of piping. My practice is to run the exhaust as direct as possible and then put the feed-water heaters on a closed branch from the exhaust pipe, so that the closing of a single valve shuts the heater off, and I find no difficulty in getting feed water from such heaters at a temperature of 200 degrees or slightly above.

On page 265 the author states the water per horse-power of the steam elevators as about 67 pounds; 100 pounds is nearer the truth.

On page 267 the author speaks of the departure from established practice in adopting 220 volts pressure instead of 110. This has been my practice in all the installations made during the last three years, and I think I have been anticipated by other engineers who

recognize the advantages from saving in copper and using the higher voltage motors.

The specifications on page 268 relating to the generators are mentioned as being rather rigid. It may be that they are for the type of generator that was purchased, but for the type of generators that I have been in the habit of using, that is the Eddy or the Western Electric generators, the requirements are very easy indeed. The subdivision into three units of uniform size is undesirable. It would have been better to have put in two 100 kilowatts and one 50 kilowatts machines, one of the 100 kilowatts machines being driven by a single engine. In this way the expense would have been decreased, the efficiency of the plant increased, and the author would have furnished us with positive information as to the real advantage of using compound engines.

The copper brushes mentioned as being used may give satisfactory service. I have tried both copper and carbon brushes, and I find that the carbon brushes are much more satisfactory in use; in one particular instance we had a 50 kilowatts generator out of which we needed to get the full capacity. We could only get 35 kilowatts with copper brushes, owing to the smallness of the commutator, small actual contact and sparking. Afterwards we put on carbon brushes and had no difficulty at all in getting the full 50 kilowatts without sparking.

In regard to the switch board mentioned on page 269, the requirements are altogether too lax. The statement that the conductors were to be proportioned so that the temperature would never rise more than forty-five degrees Fahr. above the surrounding air, and the lack of a requirement in regard to the character of the switches, the current density at points of contact where joints are to be made, and the lack of mention of the exclusion of anything but copper in the circuit, are all important features in any installation, and should not have been omitted. My own practice is to design my own switchboards, on which I use switches of my own design, in which the number of joints is cut down to one-quarter of that in the ordinary practice. Nothing but copper is in the electric circuit, and the current density at points of contact where joints are made, is reduced to 40 amperes per square inch as a maximum, and in the switches of 250 amperes carrying capacity, the current density is down to 25 amperes per square inch. On these switch boards the rise in temperature above the temperature of the surrounding air is so small that it requires a

person of very sensitive touch to be able to tell which of the conductors that are of bare copper bars have current on them and which have not. There is just an appreciable difference in temperature between the two, and no more. The losses which occur due to bad contacts and the heating of bus bars are quite considerable, and in a carefully designed plant they should be carefully looked after.

On page 270 the author mentions that "the fuel-saving would have been small, as the exhaust is used for heating during the five months of cold weather, when the load on the engines is also a maximum." In New York City, in the plants that I have charge of or have designed, we calculate on a steam service of at least six and a half months, and it would appear that he meant seven months instead of five, for the St. Louis temperature is lower than that of New York, and consequently the months of steam service should be greater.

I imagine that very few engine builders will accept the clause on page 271, in which it is stated that the crank pin is to be of the same dimensions as the shaft. All of the engines I have been purchasing for the past few years have had the crank pin of a larger diameter than the shaft. It is a small detail, but rather an important one in securing satisfactory service from the engine.

In Section E, pages 271 and 272, almost the entire page could have been omitted by the substitution of a clause that the underwriters' requirements were to be complied with, since almost all the points mentioned here are covered very carefully in the underwriters' rules.

On page 274 the author mentions the method of ascertaining the heat losses. I find that I cannot compute in that way. I, to be sure, use the units given by Mr. Wolff, which are the units adopted by the German Government; but I find it necessary to make separate computations for each floor, owing to the differing thicknesses of the wall, the heat radiation into the ground in the basement and into the air on the roof. So instead of a short formula, which can be applied to the entire building, I find that my computation books, which are about 9 inches by 12 inches, have seven or eight pages filled with the results of computations for a single building of this size.

The question, I think, could be advantageously raised as to the economy of using a steam house pump as mentioned on page 280. For some years past I have abandoned it entirely, and, so far as I

could see, with a considerable resulting economy. In an installation of this kind, where there is during a certain portion of the day but a small lighting load, and consequently the entire load is on the elevator, the additional load on the generators due to the house pump would serve to steady the voltage.

With the exception of the price mentioned for the steam-heating work, I find that the prices that the author quotes are not exceptionally low; but that on a considerably smaller installation that I am now making, that is where I am using 30 kilowatts units instead of 75, the prices are as low or lower than those he quotes. In the steam-heating system, however, it is probable that the Paul Company struck off a considerable margin of cost in order to get the contract, for advertising purposes. The elevator prices are not at all low. I think that Mr. Bryan could probably have saved ten or fifteen per cent., judging from prices that I have had recently quoted here in New York for elevators of equal grade, but higher speed and higher duty, and from the fact that the machines installed exceeded the contract requirements by a large percentage.

The figures, as given on pages 293, 294, and 295, are especially interesting as showing the difference between promise and performance. We find the test duty very high, the kilowatts hours per car mile, the coal per kilowatts hour, the coal per indicated horsepower, and so on—all very low. But when we compare from the owner's point of view the number of pounds of coal burned in the building with the number of pounds of coal burned in similar buildings, we find that the apparent economy has disappeared.

On page 293 the author says: "In spite of this, however, the improved apparatus in the new plant has kept the coal consumption down to only about 25 per cent. more than in the old building, although the work done has more than doubled." In 1894 the American Book Company employed me to advise with them in regard to the installation of their new plant to go in their new building—the New York University building on Washington Square. They were manufacturing in three separate factories, from each of which I obtained records of coal consumption, water consumption, number of men employed, etc. To manufacture 14,000 books per day, there was a fuel cost of $45. In the new plant we are turning out about 20,000 books a day, with a fuel cost of about $8.50, and in addition we are handling all of the stock through nine stories, storing and shipping that stock;

serving the New York office on the first floor, the general offices of the company on the second floor; running a high-speed hydraulic elevator service to three other floors of the New York University, and lighting and heating the entire building, in a most exposed situation.

To determine the question of economy, I sent to Mr. Bryan a telegram asking what the coal consumption per hour was; he answered: "In July, 750 pounds common coal per hour; November, 1,000;" subsequently he wrote that he considered 750 pounds as the equivalent of 600 pounds of our anthracite buckwheat. The following table, with the explanation, will show the result in comparison with some of my plants in the East:

		Duty.			
Name.	Cubic.	Lights.	Motors.	Elevators.	Coal Per. Working Hr.
Commerce Realty.....	2,315,600	2469	1-¼ H. P.	2-1500 x 300	[750] lbs.
				2|1500 x 225	600 "
				4-1800 x 150	
American Book Co....	2,180,000	1830	280 H. P.	2-3000 x 400	
				3-2500 x 250	503 "
Temple.....	2,170,000	3000	45 H. P.	2-3000 x 400	359 "
DeCourcy Bldg.....	483,000	300	64 H. P.	3-2500 x 250	
				1-2000 x 200	172 "

The Commerce Realty is the building described in the paper. The coal consumption in brackets is as stated to me in a telegram from Mr. Bryan. The figure 600 is Bryan's estimate of the equivalent coal consumption in anthracite buckwheat.

The American Book Company: general offices, storerooms, printing and book-binding establishment; output 20,000 books per day; the two high-speed elevators are hydraulic, others electric; there are two sidewalk lifts in addition; hydraulic elevators installed because of owner's requirement. This was the pioneer electric transmission plant in a book manufacturing establishment. The duty of the elevators is very severe, since the hydraulic machines make the first landing at the eighth story, and are in constant demand, and the electric machines are used for passenger and freight service constantly. I should think that the duty in useful work done was fully double that demanded by the Commerce Realty. In addition to the power development, live steam is furnished to the paste-making kettles, through a pipe over 100 feet long, the amount required being unknown. There are four dozen glue kettles kept at a temperature of about 180 degrees, by means of electric stoves.

The Temple Building, occupied in part by offices and in part by lodge rooms for Odd Fellows' meetings. Elevators hydraulic, running a large number of car miles and always heavily loaded. The building is not well lit naturally, and there are in consequence never less than 200 lights burning.

The DeCourcy Building is a manufacturing building, and in addition to the

power consumption indicated by the plant, live steam is furnished through an inch and a half pipe for paste making, steaming caps, boiling water, etc., that is in constant use. The weekly record of pounds of coal per hour and kilowatts hours of work done shows but a very slight relation. The output of the engines varies from 25 per cent. to 125 per cent. of the rated capacity, the average fluctuations being about 25 per cent., and occurring almost continuously. Practically a little live steam is required for heating in severe weather. The hours of service are very long, running 337 to 423 hours per month. The coal consumption for the past year has been 405 tons.

All of the above hydraulic elevators are operated by compound duplex pumps. The engines are all simple high speed, operated at pressures from 80 to 100 pounds. The coal burned is anthracite buckwheat. It is to be borne in mind that the fluctuations in load are instantaneous, so that while the engines rarely make three successive revolutions at the same cut-off, the demand for steam from the boilers is practically uniform, and it is possible to operate them, firing at a practically uniform rate. The elevator capacity is expressed in pounds and feet per minute, 1,500 x 300 being read 1,500 pounds, at the rate of 300 feet per minute. The plant under discussion has not been yet tested for heating, and so no comparison including heating is yet practicable.

In the case of the American Book Company, before ground was broken for the building, in fact before the plans were completed, I presented to the company six alternative lay-outs, with estimates of cost of installation and operation, prepared entirely by myself, one of which was adopted and executed. The correctness of my estimates has been practically proved by the working results of nearly three years.

The table presents very forcibly to my mind the question as to the real economy of any of the so-called economical engines for use under the conditions of a building with a fluctuating load, due either to elevators or to any other condition of service, excluding the sky-scraping office building and the mill building. For the purposes of a commercial building or a manufacturing building or any building where there is a constantly fluctuating load, such as there is of necessity where separate machines are employed, it seems to me that the experience had is conclusive against the high-duty engine. Others more familiar, perhaps, with the design of engines, can offer a more correct explanation of it than I, but it seems to me due to the fact that there must be excessive cylinder condensation.

Mr. C. R. Pratt.—Mr. Chairman, I would like to introduce my friend, Mr. George H. Hill, the Chief Engineer of the Elevator Department of the Sprague Electric Company, but not a member of this Society, who installed these elevators at the Commerce Realty building, as Mr. Hill has at hand complete data of the

history and performance of this plant, and may be able to correct some erroneous impressions created by the last speaker.

Mr. George H. Hill.—Mr. Bryan's paper has been so very satisfactory, as far as electric elevators are concerned, which is the topic of interest to me, that perhaps any further remarks of mine are not necessary. But I would like to call attention to a few points which might be of interest. The plant which has been discussed is entirely a commercial one, selected for commercial purposes, and not intended to show any abnormally high efficiency or remarkable performance of the elevators. In fact it includes a number of conditions which are rather adverse to a high showing of the elevators. For instance, the cars are very large, being 8 feet x 8 feet, or a square-foot area of 64. The speeds are low, as had been mentioned. The rails are of wood, which introduces more friction than steel rails, and the cars are corner post, and in all but three of the machines there are an extra number of idlers. The machines could not be placed directly at the edge of the hoistway, and the machine of which the test is given had a double set of idlers, thus increasing the friction losses. At the same time, with all these disadvantages, the efficiency of the machine, as a hoisting machine, is extremely high. This can be seen from Fig. 110, which gives the current consumption of the freight machine which was tested. This machine, as will be seen, carried a load of 5,000 pounds, and although not stated, the over counterbalance on this car was 2,000 pounds. This gives a net load carried on the ropes of 3,000 pounds, at a speed of 153 feet a minute, which represents 14 horse-power of actual work done. The power absorbed is seventy amperes at 220 volts, or a horsepower of twenty, giving a total efficiency over the whole machine of 70 per cent., which I think everybody will recognize as an extremely high figure for an elevator machine. This includes the efficiency of the motor, the efficiency of the worm gear and of the friction losses, which in this case, as mentioned before, are rather high, on account of the double idlers, corner wooden rails, and large cars.

There is another point which might be mentioned, and which has been brought into prominence in the table given on the board, namely, that the duty of those elevators appears small compared to the other plants. Those requirements were the altered specifications of the engineer, and while they may have affected some of the proposals, they did not the particular ones which

were accepted, and the machines installed, as the tests show, are very much in excess of the duties given. For instance, the first two machines are run at 300 feet per minute, and carried an actual load of 2,500 pounds under test. The other two machines carried, instead of 1,500 pounds at 225 feet per minute, 3,000 pounds at 256 feet per minute, and the same machine 4,000 pounds at 150. The other, rated here at 1,800 pounds at 150 feet, carried 5,000 pounds at 150 feet. This will bring the duty of the elevators more than 100 per cent. above what is shown there. The author also attributes the high performance to his reduction of requirements in the specifications, which you will note were not reduced, but are really above this *original* specification by a considerable amount, and 100 per cent. above his later requirements. This destroys this item in that clause, and places more emphasis upon the efficiency of the machine.

There is another point to which attention has been called before, but which I would like to emphasize, namely, that the rating which is customary in elevator service, and which is adopted by Mr. Bryan, of kilowatts per car mile, is not at all a fair one. In order to compare elevator machines on a rating of kilowatts per car mile, it is necessary to take into consideration the speed and the load and other conditions. It is easily possible for an elevator machine, delivering precisely the same amount of power and absorbing the same amount of power, to vary as much as 100 or 200 per cent. in the kilowatts per car mile. This is a question entirely of speed and load. If we have a machine which will carry 5,000 pounds at 150 feet per minute, it will absorb a certain amount of power, and if we double the diameter of the drum, we can carry, with the same amount of power identically, a load of 2,500 pounds at 300 feet a minute. These are both operative conditions. In the second place we will have the same amount of kilowatts consumed and delivered, but we have double the amount of mileage travelled, so that the kilowatts per car mile would be just one-half of the other.

This plant presents some rather interesting possibilities of the flexibility as well as the economy of the electric elevator. There are two machines, No. 3 and No. 4, which are through-service freight machines, carrying loads down only from the top floor to the bottom. By careful counterbalancing, and in combination with an efficient gear, which is a very important item in elevator construction of this type, the actual current consumed under

regular service by No. 3 machine was, up, five amperes. Coming down, with the average load under service, it generated into the line a current of fifteen amperes, which makes a net gain in the use of the elevator of ten amperes. In other words, it not only costs nothing to operate these two elevators, but they are actually delivering to the line a current of ten amperes each, or two kilowatts of energy, which is used in other portions of the plant. They are converting into useful and used energy the potential energy of the freight which is stored on the top floor, and is represented in the current consumption of the other elevators of the plant. A single test, of course, of these machines would represent more than 100 per cent. efficiency.

Mr. Jesse M. Smith.—How did the freight get to the top floor?

Mr. Hill.—By the other elevators. We have here a test of the other elevators, in which is given the current consumption and the efficiency for a single machine, but that does not represent the efficiency of the plant, because a portion of the energy is given back by the other elevator to the plant, to the power system.

Mention has been made of the size of generators as having been decided or affected by the elevator plant. The figures given show that they should have had very little to do with it, and I think that this can be accounted for more in the author's statement that he was providing for a very much less efficient elevator plant than he received, and for an increase in his power requirements; as it is shown, the average horse-power consumed by the elevator plant is less than 20, and as the tests show it is possible for each individual machine to start in five seconds with 50 per cent. over its running current; that is, each individual machine, based on the performance of the total power required, would be less than 30 horse-power, or 22.4 kilowatts, which is provided for by the 75-kilowatt machine; and taken altogether, taking the plant as a whole, the excess current would be very much less than 50 per cent. even—probably 40. As a matter of fact they are not starting at five seconds' start, as that is found to be too slow, but they are started in from two and a half to three seconds, thus increasing the excess current; but it is even then not more than 80 per cent. on each individual machine, and probably in the neighborhood of 50 for the total plant. So that the total starting and maximum current requirements bring the power required to about

22.4 kilowatts instead of 75. The determination of the generation therefore did not depend on the elevator plant.

The cost of repairs on electric elevators is a problem which is usually considered a great drawback. The records which we have—I speak particularly of the high class electric elevator work—are, of course, from plants installed some time ago, and as it is understood that the electric elevator is not yet very old, it is natural that portions of the apparatus have not been as well designed to stand the wear and tear on these plants upon which the data are now coming in, as they are now. I do not think this is at all a large item, nor will be a larger item than any of the other services. I do not acknowledge that it has been excessive. But at the same time it has been much larger than it will be when the various small parts of the elevator mechanism, which are liable to give way under hard service, are properly designed to meet their duty. It is purely a matter of new machine, being perfected by experience.

Another consideration in this plant, and one which must always be given prominence, is the flexibility of the system—the possibilities of it. It is possible, with an electric elevator, to have more convenience in arrangement than in a hydraulic or any other type. For instance, with this plant we have two elevators, controlled by push-button systems, which contain a single push-button at each floor, and have all the automatics of what we call our "automatic system," which is used for house service. The others are operated by the double automatic system, which contains the two push-buttons on each floor, and by which the elevator can be sent to, or brought from, any other floor. The dumb waiter in the building is operated by a box on each of the floors, by which it can be sent to any other floor; sent where desired, stopped where desired, and can be "locked out" at a floor. In addition, each machine contains all the other safeties of the normal electric elevator.

Mr. George Hill.—I would like just a moment to comment on a few of the remarks made by the previous speaker.

1. The question arises, if the elevators are so exceptionally economical, and if the engines are so exceptionally economical, whence comes the enormous coal consumption?

2. If the Sprague Company gave Mr. Bryan so much more than he offered to buy, would not the cost have been lower if they had furnished the capacity asked for?

3. Must not of necessity the elevating apparatus be inefficient, since, if a machine will lift 5,000 pounds, at 150 feet per minute, and its average duty is 1,800 pounds, that machine is certainly not working as economically as one that is designed to lift 1,800 pounds, at 150 feet per minute? If it is counterbalanced for 1,800 pounds, and is loaded for 5,000 pounds, it will not work efficiently. If it is designed for 5,000 pounds, and is counterbalanced for 1,800 pounds, it will not work efficiently. So that the elevator plant in operation, from the owner's point of view, is not a good one, not because it is electric, but because it is an electric elevator plant that is installed improperly for the duty that it is designed to perform. The difficulty with this particular form of elevator lies in the enormous starting current, which will not show in test running, but in continued starting and stopping; that is, in commercial operation. I have tested, I think, every make of electric elevator on the market, and have been watching their operation for the last eight years. I considered, in 1894, that the question of using an electric elevator in a manufacturing building was not open to discussion, and nearly three years of operation at the American Book Company proves it, for there the electric elevators were installed under the most strenuous opposition, and the result is, that on our Bristol recording gauge card for power, recording the kilowatt hours, it is practically impossible to separate the current required for the elevators from the current required for the operation of the motors throughout the building. On the ammeter you can see where an elevator is thrown on, generally—not always, but generally—because there we have a 110-volt plant, and, consequently, the amperes for these high capacity machines are large, but the actual cost of operating these elevators is practically negligible, compared with the cost of operating manufacturing machinery and the lights. The elevator load just fills in the gaps.

In regard to maintenance, there is only one make that I know of that has to apologize for a large repair account.

Mr. Bolton.—Might I add one word on the subject of the chimney, page 260? It has rather an unusual feature, I think. Inside a square brick shaft a riveted steel chimney was built, and then lined its whole length with firebrick. It seems to me that is wholly unusual and unnecessary. If it had been lined a short distance it would have been equally efficacious, and much easier to repair, too.

Mr. Kent.—Mr. Bryan not being here, I hope he will answer, in his written reply to the discussion, that question as to why there is such a large coal consumption. A possible explanation of it is that he is using St. Louis coal. That is a very different thing from the anthracite and the semi-bituminous coal used in the East. It is an especially poor coal for variable loads. If you put a lot of that coal in a furnace, nearly half of it will volatize, whether any air is supplied to it or not, or whether you want any work from the boiler or not. When anthracite coal is used, you can shut the draught off and save the coal, but the highly volatile Western coals, when the draught is shut off, continue to distil gaseous fuel, which escapes up the chimney unburned.

Mr. George H. Hill.—Another advantage of electric elevators is the fact that they are efficient at light loads just the same as at heavy loads, so that the point that the large machine is not efficient at high loads, does not affect this.

Mr. Wm. Wallace Christie.—One feature of a water-tube boiler, for use in a modern commercial building or any other, where it is located inside of and underneath the floors filled at times with a crowd of people, is that the results from possible explosion are, as a rule, not nearly as disastrous as in the old horizontal, tubular, or some other types.

*Mr. Bryan.**—The author is gratified that this paper seems to have been of such timely interest, and that it brought out so much valuable discussion. He does not claim that the plant described is the best in all respects, but in his opinion it is the best that could be secured for the local conditions, considering all the circumstances.

It will be unnecessary to answer those who thought the paper too elaborate in some details, nor those, on the other hand, who thought it incomplete. Neither need much time be given to those who have not studied the paper carefully—as, for instance, Mr. Bolton, who finds imaginary errors in the table on page 294, and Mr. George Hill, who overlooks the fact that the water rate per horse-power hour for steam elevators is stated at 100 pounds on page 255, as well as Messrs. Bolton and Christie, who think that due weight was not given to the increased safety of the water-tube boiler.

It is interesting to note Mr. Mansfield's remarkable result of

* Author's closure, under the Rules.

20.4 pounds of water per indicated horse-power hour for a 180 horse-power compound non-condensing engine. This is a higher result, however, than can ordinarily be secured, even under favorable conditions, and is better than engine builders will guarantee. The engines described in the paper would have done better had they been experimented upon further to secure the best conditions of adjustment and service, but would not have reached Mr. Mansfield's figure.

Answering Mr. Nagle—the provision that the engine should regulate with steam pressures fluctuating between 100 and 125 pounds, was intended simply to insure good engine performance under extreme conditions. As a matter of fact, there is no difficulty in maintaining the steam pressure within five pounds. An unfortunate experience with damper regulators some years ago gave me a rather unfavorable opinion of them, which no doubt the later and better makes would remove. Very few are in use in this part of the country, as their value does not seem to have been fully demonstrated. The objection to their use on down-draught furnaces, due to smoking when closed tightly, is well stated by Mr. Nagle, as is also the remedy, viz.: making the dampers fit loosely, so that even when closed, sufficient gas may escape to prevent smoking, closing the fire doors only on rare occasions, when it may be desired to stop the flow of gases absolutely. I have found no difficulty, however, in carrying uniform steam pressure by proper adjustment of the fire doors alone.

I cannot agree with Mr. Bolton that the heating system is of primary importance in New York, or in any other city. It is of equal importance with the elevator and lighting systems, but no more. An unsatisfactory lighting or elevator plant would be quite as fatal to the commercial success of a building as an insufficient heating system. The best results can only be secured by giving each of these features due study.

Neither can I agree with Mr. Bolton that in buildings of the type under consideration more steam will be required for heating than for lighting and elevator service, and that the boiler capacity is therefore determined by the requirements for heating. That this is not the case is amply shown in the paper. The boiler requirements for the most economical plant for elevators and lighting are shown to be 303 horse-power (page 256), and for heating under the severest conditions, 249 horse-power (page 279). These computations have been amply borne out by this winter's

experience. It is further borne out by records in my possession a large number of similar buildings in St. Louis, and should be still truer in New York, where the amount of heating is less—and particularly in buildings whose lighting and elevator plants are less economical than those referred to in the paper. In a different type of building, of course, with limited elevator service and less need of artificial lighting, or one whose exposure was particularly severe, the situation would be changed.

I know nothing of the computations by which the Paul people arrived at the amount of heating surface, but I do know that their guarantee covered the same external and indoor temperatures, exposures, and air changes as my original design. The system as installed has handled the building satisfactorily this winter, so much so that the owners have not thought it necessary to go to the expense of an exhaustive test. The ratio of heating surface to cubic feet of space—1 to 128—is large, but it seems to have been substantiated in practice. The representatives of the Webster system guaranteed to do the work with even less surface. There is no doubt in my mind that either system will quicken and increase the circulation. The building under discussion is well located as to exposure, and to judge the radiating surface by mere ratio to cubic feet, without taking modifying circumstances into due consideration, is quite as likely to be misleading in over as in underestimating. I am quite sure that the ratio of 1 to 128 would be permitted even in New York City, if careful and intelligent computations showed that amount to be sufficient.

In view of the fact that the Paul and Webster people very properly refuse to divulge the details of their plans and formulæ, it is not clear how any consulting engineer can design or assume responsibility for their work. It is certainly proper where a special device or system is employed, under guarantee, that the contractor for that system should be left free as to details. This is quite a different proposition from that of designing an ordinary heating plant, where my invariable practice is to work up every detail and assume all responsibility. I know of no better plan for securing bids on work of this kind than that which was pursued in this case, and which has been my practice for some time. I first design a system which will do the desired work with the minimum back pressure. Bidders are asked to submit proposals on this system, and also on the Paul and Webster systems, with permission to

reduce the radiating surface and make such other changes as they think safe for doing the same work. By permitting them to do this, they are able to utilize the benefits of their system, and at the same time reduce the cost. A large part of Mr. Bolton's argument falls to the ground when it is remembered that the proper external temperature was minus 10 degrees. It is unwise to invest money in a plant large enough to do the work at any lower temperature. On those occasions, which are rare and of short duration, we could either carry a slightly increased pressure or permit a lower indoor temperature.

Mr. Bolton is particularly fortunate in his clients, who seem willing to put in an unlimited investment in machinery. We are not so well situated in the wild and untutored West.

Neither have we been able to secure uniform proposals on water-tube boilers. In every letting we have found wide extremes of prices, in spite of the fact that the specifications were very definite. We always state the draught, character of coal, and dimensions and location of chimney, but do not limit either the grate or heating surface or rate of burning, but leave these to the boiler builder who is responsible for results.

It is to be hoped that Mr. Bolton will give the Society the benefit of his elevator researches at an early date. The competition on hydraulic elevators in the building under discussion was limited to the high-pressure type, on account of their great efficiency, due principally to fewer sheaves and less quantity of water to be handled. The high-pressure system is in very satisfactory use in a number of buildings in this city, besides the Cupples. The low-pressure hydraulic system has some advantages, but is quite wasteful in the use of fuel. The use of an accumulator of proper dimensions makes the work fairly uniform. All the results in table on page 291 were carefully secured by test, and are to be relied upon within the limits stated.

Mr. Bolton misunderstands the statements as to guaranteed and computed efficiencies. The actual water rate per indicated horse-power hour was 26.79 instead of 25, as guaranteed, and the boiler evaporation was slightly under the 7.64 guaranteed. Both the guaranteed figures are well within the limits of good practice, however, and no doubt could have been secured in the present plant with proper adjustment. Furthermore, the elevator plant was in no way responsible for the fact that the boiler and engine performance happened to fall short. A slight increase should have

been allowed for the condensation in the piping system between boilers and engines, but this was very small. A liberal allowance, however, for all these features would still leave plant C the most efficient. The three tests are strictly comparable, as they were made independently and under special test conditions. Careful records, covering the month of December, 1898, show the actual consumption in plant C of inferior coal per horse-power hour to have been very close to $3\frac{1}{2}$ pounds. On this basis the pounds coal consumed per car mile are actually 18.45 instead of 17.33, a difference of about 6 per cent., a remarkably close agreement with the computed results. It is of course true that if the engines had been driving the elevators alone, their actual economy would have been less, but in the present instance they have a large and almost constant load of electric lights, as is shown in the accompanying chart, Fig. 112. The fluctuations are therefore a smaller percentage of the total, and the load is fairly uniform in spite of the extreme fluctuations due to the elevators. An important advantage of the electric system is that by combining it with the lighting, the fluctuations which are so detrimental to economy can be largely reduced.

The hydraulic system of the Bowling Green building no doubt remedies the losses due to carrying the excessive pressures which are only required for occasional heavy loads, but as a double system of pumping is necessary, it would seem that the reduced pump efficiency would go far towards offsetting that gain.

Our reasons for selecting 75 kilowatts units instead of smaller ones—which Mr. Bolton does not approve—are stated on page 266, and have been abundantly substantiated in the working of the plant. While the lighting load could have been carried with two smaller generators, no provision would have been made for growth, and an accident to one unit would have seriously crippled the plant. The size of units was selected independently of the elevator problem, a third unit of the same size being added on account of the elevators. This unit, as has been shown, is really larger than is needed, and is certainly all that should be included in the cost of the elevator plant.

Our experience with electric elevators does not coincide with Mr. Bolton's. We have not found them luxuries, which "come high and must be paid for." We have found them quite able to take care of themselves on a strictly business basis. This must

not be construed to mean that we always prefer the electric elevator. There are other conditions of service where the hydraulic, or even the steam elevator, may be preferable. It is true that the electric elevator has its reputation still to make as to reliability and low cost of repairs, but the latest and best types are meeting all reasonable expectations in these respects, and it may be assumed that further improvements will be made as experience shows them necessary.

If Mr. Bolton will examine the amount and hours of electrical load in the building under discussion (Fig. 111) he will, perhaps, agree that the selection of compound engines was not unwise. I am surprised that he should assume that the simple engine will give better results with variable loads, as the fact that the relative economy between simple and compound engines is approximately the same with fluctuating as with steady loads, has now been well established. I have not found that compound engines are less reliable, nor that they increase the labor.

We will all agree with Mr. George Hill's statement that all the engineering work of a building ought to be done by one engineer. But is such a thing possible? If Mr. Hill—who is not only an engineer, but an architect—is also an expert in plumbing, sanitary and structural work, and the mechanical and electrical systems, which cover the boilers, furnaces, engines, heating, ventilating, lighting, elevator and hydraulic plants, and is able to keep abreast with the best practice in all these widely different lines, he may well be termed one of the most remarkable men of the century. My own observation and experience lead to the belief that when a single person attempts to cover so broad a field, the covering is likely to be a little "thin, in spots." The consensus of opinion of the present day is unquestionably that the best results are attainable only by specialists in these several lines.

It is proper to say that I am indebted to my business associate, Mr. H. H. Humphrey, member American Institute of Electrical Engineers, for assistance and data regarding the electric plant of the building under discussion, this part of the work having been entirely in his hands. It seems to me that the only method by which the various engineering features of a single building can be made entirely harmonious is to have them handled by a firm whose members are specialists in the different lines, or who have specialists in their employ.

Mr. Hill is also fortunate in that he is able to make up his mind

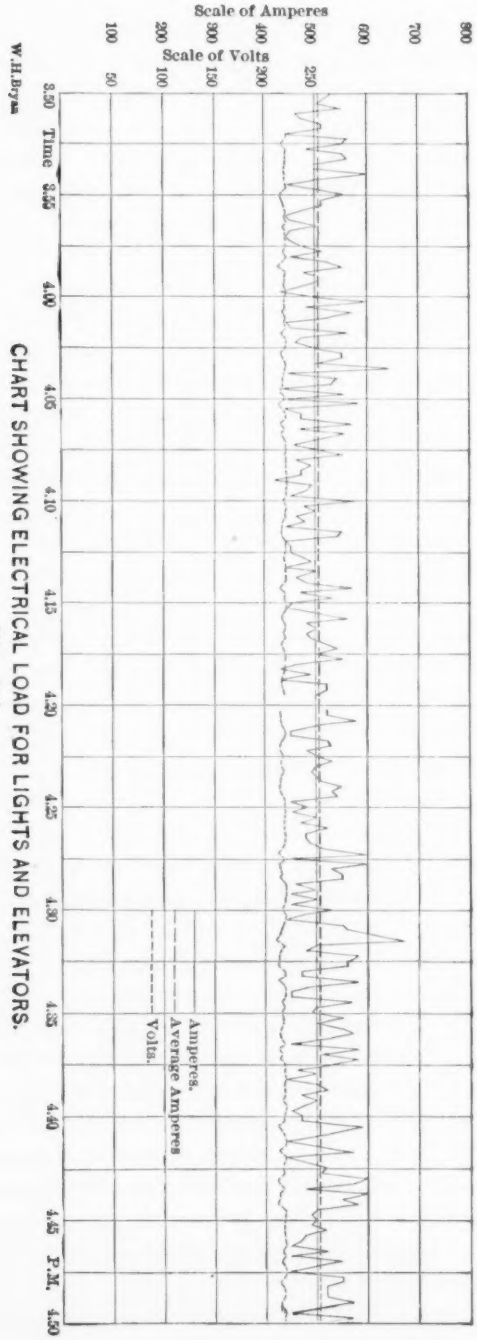


CHART SHOWING ELECTRICAL LOAD FOR LIGHTS AND ELEVATORS.

Fig. 111.

in advance as to just what he wants in any particular building, and thus avoid the necessity of asking for alternative bids. In our experience, improvements are being made so rapidly in certain branches—such as the elevator and electrical installations—that the progressive engineer must thoroughly investigate new methods, however much he may be inclined to be conservative. Our experience shows, also, that competition is secured by getting bids on a number of different types, as well as different bids on the same type. It is not clear how Mr. Hill would get close or competitive figures by drawing his specifications so as to fit only one system. Furthermore, the cost itself is always an important element in making a selection, and this cannot always be determined in advance, even approximately. While the client's best interests could be served by confining the bidding to a single specification on some parts of the work, it would be unwise to do so on others.

As to excessive elevator requirements, these were stipulated by the tenants, and were objected to by the engineers at the time, but without success.

Mr. Hill seems, also, to have been particularly fortunate in having designed such a large number and wide variety of buildings as to be able to determine boiler horse-power by mere comparison with a similar building previously designed. Our own experience has been that one building forms no precedent for another, except in the most general way. The only safe plan is to study each building independently and on its own merits. Buildings of approximately the same cubic contents, and for the same general class of business, may differ widely in their requirements for steam.

The indirectness of the piping system criticised by Mr. Hill is due to the necessity for avoiding interference with the smoke flue, the head-room being limited (see Fig. 106). The steam pipe was enlarged to allow for this increased length. The pipes were so large, and the engines so close to the boilers, that no receiver was deemed necessary. The large number of fittings which Mr. Hill criticises on the exhaust is due to the necessity of fitting the plant into the available space to best advantage. I have not found it satisfactory to connect exhaust heaters on closed branches. It will be noticed that the heater outlets serve, also, as connections to the heating system, and the double connection certainly improves the efficiency of the heaters as expansion tanks.

If we are to understand that Mr. Hill invariably uses 220 volts

pressure instead of 110, he is not in harmony with the best practice of the day. This depends upon many features, and 110 volts will probably continue to have the preference for ordinary isolated work, particularly as the latest forms of motors require greatly reduced starting currents.

Mr. Hill thinks 2–100 kilowatts and 1–50 kilowatts unit would have been better than 3–75 kilowatts, one of the 100's to have a non-compound engine. We do not agree with him. Theoretically the three units should be different in capacity, so that a large number of combinations may be made in order to suit the different loads, the efficiency being better at or near rating. In practice, however, this does not work out, as the operatives will not change engines over for small variations of load. Furthermore, the efficiency curve is reasonably level between 75 per cent. and 120 per cent. of rating, and only drops seriously outside of these limits. On the other hand, the advantages of units that are identical are many. They not only cost less, but they usually fit the space available better, and only one set of repair parts, packing, etc., need be carried. They are absolutely interchangeable, and an accident to one of them impairs the service no more than another. The only case where a small unit is justified is where it is known that there will be a light load for long hours. The accompanying statements of actual load show that the selection was amply justified in the present instance.

As to the switchboard, the specifications mapped out the design in a general way, but did not go into minute detail, the contractor being required to submit his detailed design for approval. When he did this, all the details were carefully checked, and such changes made as were found necessary. It is desirable to use standard apparatus, as far as possible, in order to keep the cost down. A special and original design of switchboard—and particularly a new design of switches, would certainly have made the cost excessive. We have had no difficulty in finding switches of satisfactory design on the market. The points which Mr. Hill mentions are all good ones, provided they are not carried to the point of unreasonable cost. It must not be understood that the bus bars were allowed to heat to the 45-degree limit, which was intended to cover joints and switch clips.

As to the months of heating in St. Louis, the coldest months are, of course, November, December, January, and February. There is quite a falling off in March, and a still greater one in

April and October. These three months together do not usually require as much heating as one of the others. It is true that some heating is done during seven months of the year, but that means four months of full heating and three months of very light service.

There is, of course, no objection to making the crank pin greater in diameter than the shaft, but it should not be less.

We cannot agree with Mr. Hill that two pages might have been omitted in Section E by simply calling for compliance with the underwriters' rules. A careful checking of those pages shows but three sentences which might have been omitted, all the rest being special. At that time the St. Louis Underwriters' rules did not fully cover 220-volt work. For instance, we specified the thicknesses of rubber for different sizes of wires, which is a point the underwriters have taken up since; also an insulation resistance test based on length of wire in circuit, instead of number of lamps connected, which we believe is better practice. We limited the number of lamps per circuit more than would have been permissible under the underwriters' rules. These rules no not specify size or drop. We always require underwriters' inspection, and their rules are good as far they go, but they do not cover all desirable features.

Each floor and each room of the building must be computed independently for the heat loss. The wall factor, however, was not changed for each floor, as the thickness varied but little, and an average value was assumed; the natural tendency of heat to find its way to the upper floors, as well as the increased amount of radiation on the top floor, being considered sufficient.

The objections to electric pumps are their greatly increased first cost, greater complication and liability to derangement. They could not be used satisfactorily for boiler feeding on account of their low motor efficiency at slow speeds. These facts, in our opinion, overbalance their better steam efficiency. An electric house pump would, of course, have given satisfaction, but we did not feel justified in recommending it on account of its increased cost, and having two different kinds of pumps in the plant.

The most important criticism which Mr. Hill makes, however, is that there seems to be a discrepancy between promise and performance, and that while the test duties are high, the actual results are poor. He mentions a number of buildings in which

the consumption of fuel is lower, although they have less efficient apparatus, all of which he submits as an argument against the use of improved apparatus. A moment's consideration, however, will show the impropriety of thus comparing buildings which are used for widely different purposes. In order to make a more equitable comparison, we have looked up the coal consumption of a number of large modern buildings in this city, which are of such a character as to be fairly comparable with the Commerce Realty Company's plant. All of them have hydraulic elevators and non-compound electric light engines. The number of tons of 2,000 pounds of soft Illinois coal consumed in these buildings per annum for all purposes for each 1,000 cubic feet of space above ground level, is as follows: 4.40, 2.96, 1.91, 1.88, 1.73, and .92, the last being the Commerce Realty Company's building. The latter is, therefore, using only a trifle over half as much coal as the best of the other buildings, and about one-fifth as much as the worst. This proves that the coal consumption is not, as Mr. Hill assumes, large, but, on the contrary, is very small indeed. For instance, during the month of December, 1898, the average load on the engines was 237.5 horse-power, of which a little less than 10 per cent. was for elevators and the balance for lighting. This large lighting load is due to the building being a very dark one. For the same month the coal consumption was a trifle under 1,000 pounds per hour, which includes coal used for banking fires and raising steam in the morning, as well as radiation and all other losses, and the heating of the building when the engines were not running. This consumption of four pounds of inferior coal per indicated horse-power hour is certainly a good result, being equivalent to about three pounds of the anthracite screenings which Mr. Hill mentions. It is also equivalent to about three and a half pounds of our coal per indicated horse-power hour while the engines were in operation. In our opinion simple engines would have required about 50 per cent. more fuel for the same work. During the week of December 19th to 25th a careful log was kept of the electrical load, and it was found to range between 400 and 800 amperes, averaging 615. There are now so much data available as to the economy of compound engines and electric elevators that these points should be no longer left open to discussion.

Mr. George H. Hill contributes some interesting data on elevators. The use of the corner guide posts, wooden rails, and increased number of idlers were matters beyond our control. Our

own preference was to locate the elevator mechanism at the top of the shafts, which we have often done with satisfactory results, but this was opposed by the contractors. The machinery would have been somewhat less accessible, but would have occupied less valuable space, and would doubtless have been more efficient. Mr. Hill's figures are slightly in error. In computing the efficiency he takes the amperes at 70 (see Fig. 111), which was the current required just before the car reached the upper limit of travel, when the cables had all gone over the sheaves and were assisting the counterbalance. This explains the reduction in power in each of the different cases. A fairer reading would be midway of the travel, at which point the amperes consumed were 75. Computations made on this basis show the hoisting efficiency of the elevators to have been not 70, but 63.6 per cent., which result is nevertheless excellent. Similar computations with 4,000 pounds load gives an efficiency of 53½ per cent., which is about what would ordinarily be expected. With a load of 1,000 pounds, the efficiency dropped to 45.6 per cent. Similar computations show that the total horse-power to lift 5,000 pounds at 153 feet per minute was approximately 22, of which about 7 was required to overcome friction, 23 to lift the useful load, and 8 was derived from the counterbalance. The above computations do not take into consideration the extra power required in starting. The efficiencies, however, are not materially above those anticipated, as is shown by comparing the table on page 264 with that on page 288.

Mr. Hill also makes an error in his definition of the unit of elevator efficiency. It is not kilowatts per car mile, but kilowatts *hours* per car mile, which takes into consideration the element of time. The efficiency thus stated is dependent upon speed, as Mr. Hill states, but not on load. Whatever the live load or the counterbalance may be, the power required to lift them is returned to the line on the return trip, less friction. Assuming, therefore, that the same total load is ultimately lowered as well as lifted, the power required is only that necessary to overcome friction, motor losses, and acceleration. Load, therefore, only comes in so far as it affects friction, motor efficiency, and starting current, so that, after all, the efficiency unit customarily used—kilowatts hours per car mile—if not exact, is by no means so greatly in error as Mr. Hill assumes. If, in using this unit, the loads, speeds and frequency of stops are started, so that their

effect on friction and acceleration may be taken into consideration, and the records are taken for round trips, it is, in our opinion, an equitable basis of comparing elevator efficiencies. We have sometimes required the efficiencies to be stated in kilowatts hours —or their equivalent—per ton mile of *live* load, but this is no better than the other, as the power consumption does not vary in direct proportion to the load. It is true, as Mr. Hill states, that the same kilowatts will be required to lift either 5,000 pounds at 150 feet, or 2,500 pounds at 300 feet, and in the latter case the kilowatts hours per car mile would be only half the former. This is due wholly to the increase of speed, however, and not at all to the load, as that equalizes itself on the down trip. Leaving starting current out of consideration, the power consumed for round trips is wholly expended in friction and motor losses. But it must not be overlooked that loads and speeds have important effects on friction, motor losses, and acceleration.

Mr. Hill is, also, certainly unwarranted in his statement that the size of the electrical generating plant is influenced but little by the elevators. It is shown in the paper that the average electric horse-power used by the elevators, over a long period of time, is about 20, which may be increased to 25 at the busiest seasons. The most serious feature, however, is the starting current, which, as shown in Fig. 110, is about 50 per cent. greater than the running current, so that a single elevator requires over 30 horse-power when starting, and if a number of them start at once—as frequently happens—100 horse-power, or more, may be thrown on the generators at one time. The latter must, therefore, not only have an increased capacity of 25 horse-power for average service, but must be able to take overloads of three or four times this amount.

The accompanying chart, Fig. 111, shows the current consumption for an hour, on the afternoon of January 25, 1899, readings being taken at 15-second intervals. Fig. 112 shows the last 20 minutes of the same hour, with readings taken at still shorter intervals. In Fig. 111 we have, also, plotted the average electrical load and the voltage, which, on the whole, tell an interesting story. These readings were taken on a bright day, and at a time when the elevator service was low, so that the results do not average as high as the work done in December. The chart points out the advantages of a large lighting load for steadying elevator service. It also clearly shows the reason of the large coal

consumption, which was such a bugbear to Mr. George Hill, and is an argument which, we think, will be conclusive to most parties as to the efficiency of the apparatus, when it is remembered that the coal consumption, for all purposes, was less than 1,000 pounds of inferior coal per hour.

A serious objection to the electrical elevator, heretofore, has been its excessive starting current. This is being reduced in the latest forms of motors, until to-day it is no longer the serious objection that it was. At the time the observations shown in Figs. 111 and 112 were taken, the elevators were adjusted for about

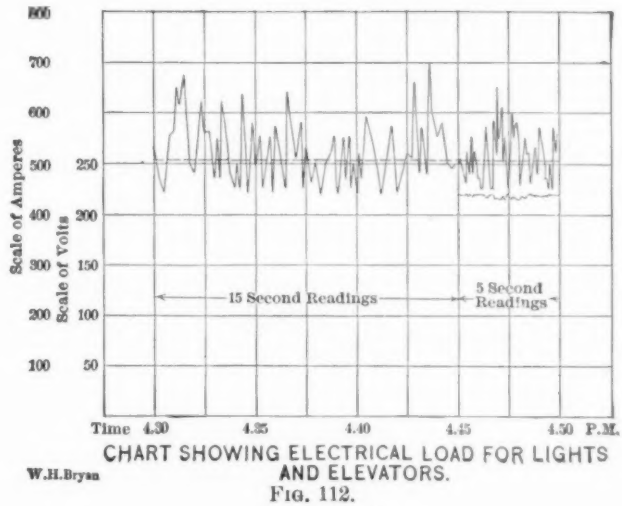


CHART SHOWING ELECTRICAL LOAD FOR LIGHTS AND ELEVATORS.

FIG. 112.

a three seconds' start. With this increased to five seconds, the fluctuations of current would be much less severe.

Mr. George Hill's first inquiry has been fully answered. Answering his second, will say, that if the Sprague Company gave more than a strict interpretation of the contract required, they probably had good reasons for doing so. Elevators following the revised specifications would certainly have cost less. Answering his third inquiry, will say, that to get the best efficiency, the service required of each elevator must be studied to ascertain its average work, and then it should be counterbalanced accordingly. A machine of the "Z" type, with pilot-motor control, can be adjusted, for efficient results, with light loads at high speeds, and, at the same time, be able to carry increased loads at reduced

speeds, and with practically equal efficiency and with the same counterbalance.

As to Mr. Bolton's criticism of the chimney being lined with fire brick throughout, will say, this was done at the desire of the architects. This is the only case where we ever did it, but we thought it would be an opportunity to compare results. It has some advantages in increasing the life of the stack, and by reducing heat loss, would maintain the draught better, but we agree with Mr. Bolton that, as a general proposition, the expense is unwarranted.

DCCCII.*

THE CALORIFIC POWER OF WEATHERED COALS.

BY R. S. HALE (ASSOCIATE MEMBER), BOSTON, AND HENRY J. WILLIAMS, BOSTON.

In 1896, samples of Maryland, Virginia, Pennsylvania, and Ohio coals were collected and arranged for testing as follows: For tests of fine coal the samples were ground in a coffee grinder, and thoroughly mixed and divided into two parts. For tests of lump coal the coals were broken into lumps of about nut size, and alternate lumps taken from the pile to form two samples.

Having thus obtained two reliable samples of each coal, or

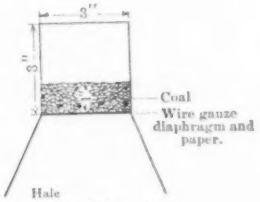


FIG. 113.

four samples, where tests of both fine and lump coal were to be made, one sample was tightly sealed in an ordinary pint fruit jar, while the corresponding sample was exposed on an uncovered balcony out of-doors for eleven months in an uncovered tin can provided with a diaphragm or bottom of fine wire gauze, as shown in the accompanying sketch (Fig. 113).

Rain and snow fell upon the coal, but the wire diaphragm permitted the water to drain off, while a paper disk placed upon the wire gauze prevented the coal from sifting through the meshes.

The lump samples were exposed in pans of much larger size, which were provided with holes to let the water drain off.

This method of exposure closely reproduces the conditions to

* Presented at the New York meeting (December, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

which the surface of a pile of coal, left out-of-doors, would be exposed. The interior of such a pile, however, would suffer less than the samples.

At the end of eleven months all the samples were analyzed by Mr. Henry J. Williams, together with a sample of Pocahontas coal that had been exposed in a coal yard for three years, and one of Cumberland coal that had been under cover for three years. These last coals, however, being different, are not comparable with each other.

The results of the above analyses are shown in the accompanying table.

Proximate analyses of the coals may be found on lines 1, 2, 3, and 4.

In these analyses the percentages of ash in some of the exposed samples are unfortunately too high, for a little gravel was accidentally washed off the roof of the house, by the rain, into some of the cans. This, however, in no way affects the relative percentages of combustible matter free from ash.

The percentages of sulphur are found on lines 5 and 6, while lines 7, 8, 9, 10, and 11 give the ultimate analyses, calculated to combustible. The British thermal units calculated from the analyses by the formula suggested by the boiler committee of the A. S. M. E., viz.: "$146C + 620(H - \frac{O}{8}) + 40S$," appear on line 12, while the British thermal units interpolated from Kent's curve, which is based upon the per cent. of fixed carbon, as found by proximate analysis, may be found on line 13.

Line 14 shows the per cent. of fixed carbon in the combustible. Lines 15 to 21 record the changes in the various quantities due to weathering, being plus for an increase, and minus for a decrease.

Line 22 gives the free hydrogen, and line 23 its increase and decrease, due to weathering.

Finally, on line 24 the calorific power of three samples of the above coals, both sealed and exposed, as determined in Mr. Williams's bomb calorimeter, is shown.

The average of the results obtained shows that weathering, under the conditions described, decreases the percentage of carbon, hydrogen, nitrogen; increases the percentage of oxygen, and does not materially alter the percentage of sulphur.

The conclusions to be drawn from an examination of the results shown are:

1st. That weathering decreases by about two per cent. the theoretical calorific power, as calculated by Dulong's formula.

2d. That weathering decreases by about one-half of one per cent. the actual or true calorific power, as shown by the three results obtained with the bomb.

3d. That if the results calculated from the above analyses are interpolated from Kent's curve, which is based on the per cent. of fixed carbon, as determined by proximate analysis, the latter being a more or less variable quantity according to the age and degree of exposure of the coal, the remarkable result is obtained that the weathering of a sample of coal *increases* the percentage of fixed carbon in the combustible, and also *increases* by about *one per cent.* (1%) the calorific power of the coal.

It may be noted that the English form of Dulong's formula, which neglects the term $\frac{O}{8}$, but uses slightly lower values for the heat of combustion of C and H, would cause the calorific power, as computed, to agree more closely with the results obtained with the bomb than the formula "$146C + 620(H - \frac{O}{8}) + 40S$." This is especially noticeable when the weathered and unweathered samples are compared.

In a general way, the results show a slight diminution of calorific power in the grades of coal examined, directly traceable to weathering. They also show that where coals have not been exposed to oxidation, by weathering or otherwise, there is a reasonably close agreement between the results calculated by Dulong's formula and those obtained with the bomb calorimeter.

Last of all, they show that when it is desired to compare the respective calorific powers of coals that have been altered to a greater or less extent by oxidation with those of coals that have not been exposed to the weather, the use of Dulong's formula, or any of its modified forms at present in use, as well as the use of Kent's curve, would lead to erroneous conclusions.

DISCUSSION.

Mr. F. W. Dean.—I happen to have analyses of samples taken from one load of picked Pocahontas coal—one taken on April 22, 1895, when the coal had been constantly under cover from the time it was mined; and the other taken on September 25, 1896, the coal having been out-of-doors during the interval

between, or seventeen months. This coal was picked at the mine and shipped in box cars. The analyses of the samples are as follows:

April 22, 1895.		*September* 25, 1896.	
Carbon	87.70	Carbon	87.11
Hydrogen	5.20	Hydrogen	4.34
Nitrogen	1.25	Nitrogen	1.32
Oxygen	2.68	Oxygen	4.06
Ash	2.49	Ash	2.55
Sulphur	0.68	Sulphur	0.62
Computed calorific value	15,860	Computed calorific value	15,026

It would probably be assuming too much to say that the coal had lost the heat value indicated by the analyses, or $3\frac{1}{2}$ per cent., for we cannot positively say that each sample represented the coal, for neither was a sample of the whole lot, but of the parts that were used on two boiler tests. Therefore, these results can merely pass for what anybody happens to think they are worth.

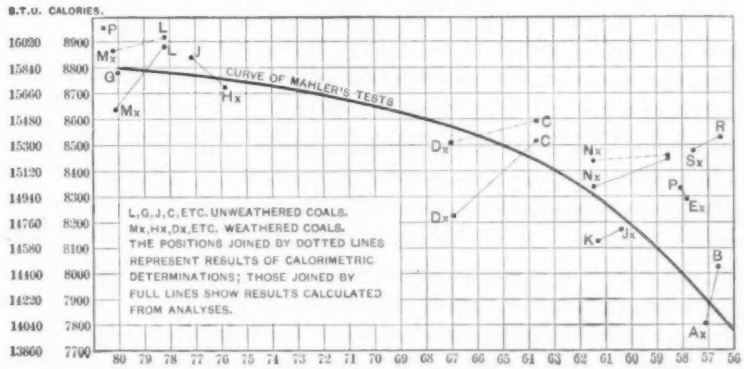
Mr. William Kent.—The results obtained by the authors of the paper are important and valuable, in showing that the loss in the heating value of bituminous and semi-bituminous coal is not as great as it is generally supposed to be, and in giving us increased confidence in the Dulong formula for calculating heating value from the ultimate analysis—provided the analysis is correct. In order to study the results given in the complete table of figures, I have made the following short table, which may be of interest:

Coals.	Fixed C. in Combustible.	Heating Value per W. Comb., B.T.U.	
		By Analysis.	By Calculation.
George's Creek, L. M.	inc. 2.4 per cent.	dec. 427	90
New River, I. H.	dec. 1.4 "	dec. 208	
Portland, O., C. D.	inc. 3.4 "	dec. 611	160
" " B. A.	inc. 0.3 "	dec. 341	
McDonald, Pa., K. J.	dec. 0.8 "	inc. 63	
" " O. N.	inc. 2.9 "	dec. 220	40
Pittsburg, Pa., P. E.	dec. 0.2 "	dec. 95	
" " R. S.	inc. 1.0 "	dec. 93	
Average	inc. 0.95 per cent.	dec. 241	97

The average decrease in heating value by weathering, according to the analyses, is 241 B.T.U. per weight of combustible,

or about 1.6 per cent., and according to the calorimeter it is (for three samples only) about 0.67.

The authors have compared their results with a curve which I plotted from the results of Mahler's calorimetric work and published in volume i. of *Mineral Industry*, 1890, and also in 1897 in volume xxiv. of the *Transactions of the American Institute of Mining Engineers*, in connection with a discussion of the



CALORIFIC POWER OF COALS, WEATHERED AND UNWEATHERED.

FIG. 114.

work of Professors Lord and Haas on American coals. As this curve has frequently been referred to by recent writers, and as it has now fairly established its validity within the limits I have claimed for it, I will ask to have its most important part reproduced with this discussion, with a slight modification which later studies have made advisable.

A table derived from the curve is as follows:

APPROXIMATE HEATING VALUE OF COALS.

Per Cent. of Fixed Carbon in Coal Dry and Free from Ash.	Heating Value.		Per Cent. of Fixed Carbon in Coal Dry and Free from Ash.	Heating Value.	
	Calories.	British Thermal Units.		Calories.	British Thermal Units.
97	8,300	14,940	63	8,400	15,120
94	8,450	15,210	60	8,200	14,760
90	8,600	15,480	57	7,900	14,220
87	8,700	15,660	55	7,700	13,860
80	8,800	15,840	53	7,400	13,320
72	8,700	15,660	51	6,900	12,420
68	8,600	15,480			

The figures in the original table published in 1890, which have been changed in the above table, are the following: 97,

8,200 calories; 94, 8,400; 60, 8,100; 57, 7,800; 54, 7,400; 51, 7,000; 50, 6,800.

The original claim made for this curve was as follows: "Knowing the percentage of fixed carbon in the dry coal free from ash, we may, in the case of all coals containing over 58 per cent. of fixed carbon, predict their heating value within a limit of error of about 3 per cent."

In 1897 a study of Lord and Haas's work led to the following statements: "Excluding the coals that have below 58 per cent. of fixed carbon in the combustible, the variation of any one of Lord and Haas's coals from the Mahler line does not exceed 320 calories (576 British thermal units), or about 4 per cent. Taking the average figure for each class of coals [there were seven classes, 40 coals in all, viz.: Pocahontas, 5; Hocking Valley, O., 5; Thacker, W. Va., 4; Pittsburg, 7; Middle Kittanning, Pa., 7; Upper Freeport, Pa. and O., 11; and Mahoning, O., 1], it falls in all cases within the limits of 3 per cent. . . . Taking into consideration the fact that the reported percentage of fixed carbon is very apt to be 2 or 3 per cent. in error, I am disposed to hold to my original conclusion, at least until a larger series of tests may show that it should be modified. It is to be observed, however, that the Mahler line falls rapidly with percentages below 62 per cent. of fixed carbon; and it is therefore to be expected that below this point there will be a greater range of variation in heating value than above it. When the volatile matter exceeds 38 per cent. an increasing proportion of it is oxygen, and the relative proportion of oxygen in the highly volatile coal varies in the coals of different districts, as is shown by Lord and Haas's analyses. Thus the Upper Freeport coal averages only 9.58 per cent. of *O* (in the coal dry and free from ash), while the Hocking Valley coal averages 16.10 per cent., although both coals have the same percentage of fixed carbon, viz., 58 per cent. Full credence, therefore, is to be given to the conclusions drawn from Lord and Haas's tests, that, when the fixed carbon is less than 62 per cent. of the combustible, each class of coal has a law of its own, and coals of any one class may differ in heating power from the coals of another class containing the same percentage of fixed carbon, to an extent as great as 5 per cent., as in the case of the Upper Freeport and the Hocking Valley coals."

Messrs. Hale and Williams in their third conclusion say: "If

the results calculated from the above analyses be interpolated from Kent's curve . . . the remarkable result is obtained that the weathering of a sample of coal increases the percentage of fixed carbon in the combustible, and also increases about 1 per cent. the calorific power of the coal."

It is not at all remarkable that the percentage of fixed carbon should increase by weathering, and this fact is learned without the use of my curve, by simply referring to the analyses. As shown in the small table given on page 336, the average increase in fixed carbon is 0.95 per cent., and an increase took place in five samples out of eight. There is no reason why an increase should not be expected, for it is at least as probable that weathering should cause an oxidation and disappearance of volatile matter as that it should cause an oxidation of fixed carbon. It is not impossible that it should cause an increase in the heating value per pound of combustible, for that would ensue if the weathering should cause the loss of that portion of the volatile matter which contains oxygen rather than hydrocarbon, or if it should cause the oxidation of fixed carbon, leaving in the coal the more valuable hydrocarbon.

It is scarcely fair to the curve to condemn it for showing an average increase of heating value of 1 per cent., instead of the decrease of 0.6 per cent. shown by the calorimeter, when the curve was not derived from weathered coal, and especially when the best claim made for the curve is that it will give results within a limit of error of 3 per cent.

On the accompanying diagram I have plotted the results obtained by Messrs. Hale and Williams, referring them to my curve. A study of the diagram shows the following:

1. All the analyses and calorimetric results on coals containing more than 59 per cent. of fixed carbon in the combustible, weathered or unweathered, are within the stated limit of error of the curve 3 per cent., with the single exception of the result calculated from the ultimate analysis of the weathered coal D. This exception is evidently due to an error in the analysis. The proximate analysis shows an increase in the fixed carbon by weathering of 3.68 per cent. (referred to combustible), while the ultimate analysis shows a decrease in the total carbon of 0.99 per cent. These figures appear incompatible.

2. The coals containing less than 59 per cent. fixed carbon show, in most cases, a wide divergence from the curve, tending to confirm

the conclusion drawn from the work of Lord and Haas that among the highly volatile coals each class of coal has a law of its own.

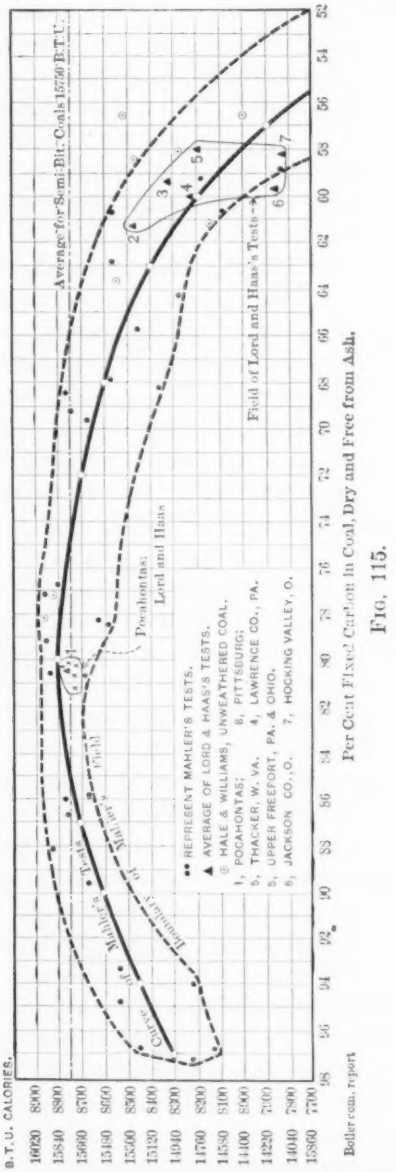


FIG. 115.

3. Coals A B and C D, both said to be Yorkville lump, from Portland, O., show such a great difference both in per-

centage of fixed carbon and in heating value that they appear to belong to entirely different classes of coal. It would be interesting to know whether these samples came from the same seam or from different seams. If from the same seam the figures would indicate that the conclusion of Professors Lord and Haas, that the coals mined from one seam over a considerable area of country have a nearly uniform heating value, has some extraordinary exceptions.

It should be noted that the loss in heating value per pound of the combustible portion of the coal may not be a true measure of the actual loss in heating value of the whole of a given lot of coal, for besides the loss in heating value per pound there may be also a loss in weight, and this, if any, expressed as a percentage, should be added to the loss in heating value per pound.

DCCCIII.*

EXPERIMENTS ON THE FLOW OF STEAM THROUGH PIPES.

BY R. C. CARPENTER, ITHACA, N. Y., AND E. C. SICKLES, ANACONDA, MONT.

THE experiments described in the following paper were made to determine the coefficient of friction of steam flowing at different velocities through pipes and fittings.

The results obtained, however, seem to be of such a nature as to warrant their use in connection with well-known formulæ for the flow of compressible fluids, and a table is submitted which gives the probable amount of steam delivered through pipes of different lengths and diameter.

One series of experiments was performed by measuring the condensation and flow through a line of piping 90 feet in length and 1, 1½, and 2 inches in diameter, which was erected in a tunnel leading from the Sibley College boiler rooms under the adjacent street.

For this purpose a pipe line was, for each test, erected in a horizontal position, and the steam before entering the line, and after leaving, was passed through a steam separator. Very accurate and carefully calibrated steam gauges with dials 14 inches in diameter, and subdivided so as to read fractions of a pound, were employed to determine the initial and final pressures, and the drop in pressure. The quality of the steam was tested by the use of a throttling calorimeter after leaving the first separator and before reaching the second. The first separator was employed for the purpose of removing all free water from the pipe, and insuring dry steam or steam with so little moisture that the quality could be accurately determined. After several trials one was obtained, which gave very satisfactory results.

On account of the large amount of steam required for the

* Presented at the New York meeting (December, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

experiment it was impracticable to furnish it in a superheated condition.

The experiments were made with the pipe uncovered, and also with the pipe very carefully covered with thick coatings of hair felt and asbestos.

A later series of experiments was made to determine the coefficient of friction of the steam discharged from a boiler plant of 525 boiler horse-power through a 3-inch pipe 95 feet in length. Finally the results were checked by referring to experiments made some years previous by Messrs. Green & Stratford to determine the friction in a pipe 195 feet in length, which was used to convey steam from the boiler plant to the engineering laboratory.

A diagram of the arrangement of piping which was used in

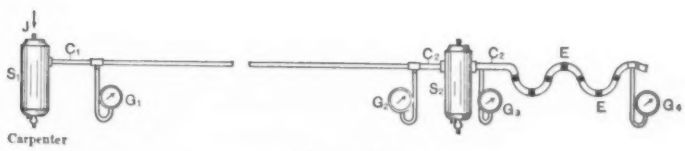


FIG. 116.

the first series of experiments is shown in Fig. 116. The weight of steam flowing in a given time was calculated from the known pressure behind discharge orifices whose constants of flow were known. This latter quantity was of use principally in determining the velocity of steam flowing through the pipes.

The Problem Involved.—The problem involved in the experiment required the measurement of the loss of energy which took place in the pipe, and the elimination from this of all losses which were not due to friction.

Where steam flows in a given pipe there are several factors which tend to dissipate or change the form of energy possessed by the steam. These may be classified as follows: (*a*) condensation, (*b*) friction, (*c*) expansion with changes of external energy, (*d*) the effect of gravity.

(*a*) The condensation may be divided into two parts, one the static condensation, that which occurs when there is no flow of steam; the other the dynamic condensation which occurs when there is a flow of steam. The latter should be less than the former on account of fall of pressure and temperature at the

delivery part of the pipe, which tends to raise the quality of the steam, but the fall in pressure also involves a change in kinetic energy, and this, with other influences, seems to have made the amount of condensation very nearly the same for both cases.

(*b*) The friction in the pipe would cause a loss of pressure and require work to be done. If the pipe were a non-conductor and the expansion adiabatic, then there would be no loss of energy, but there would be a transformation of initial potential energy into external and kinetic. This change of energy between two points could be equated to a form involving the coefficient of friction, and its value thus derived. In practical work the adiabatic condition is seldom or never realized, for a pipe may be covered ever so well and still there is a loss of heat.

(*c*) The expansion would cause change in external latent energy which is accounted for by the steam table under different absolute pressures.

(*d*) The effect of gravity may be considered as *nil* in this series of experiments, as the pipe was horizontal. If the pipe were inclined any perceptible amount from the horizontal, the work could be computed, due to the delivery, and the height through which the steam raised or fell. We shall consider only the case of a straight, horizontal pipe.

General Equations.

If P be any pipe of uniform diameter, d, and E_1 the energy of the steam entering a section, as A, in one second, and E_2 the energy leaving a section B, at the distance of L feet from A, then

$$E_1 - E_2 = E_c + E_f + E_e + E_g \quad . \quad . \quad . \quad . \quad (1)$$

where E_c = dynamic condensation energy.
E_f = friction energy.
E_e = expansion energy.
E_g = gravitation energy.

Now, since in steam tables the energy E_e of the latent external energy is included in the total energy of steam at any pressure, it need not be considered for our present purpose; also E_g is negligible because our pipe is horizontal.

Writing E'_1 and E'_2 for the new form of (1), we have:

$$E'_1 - E'_2 = E_c + E_f \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

This is an indeterminate equation for certain values of E_c and E_f, but if E_c can be determined by direct measurement, or if the pipe can be so protected that the condensation loss is negligible, the equation becomes determinate. It is also possible to make the velocity of steam so much that the heat developed by friction shall make the loss by condensation zero, in which case the equation is also determinate. Let the total loss of energy be denoted by E, then we have that

$$E = E'_1 - E'_2 = E_c + E_f \quad . \quad . \quad . \quad . \quad . \quad (3)$$

from which

$$E - E_c = E_f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Various expressions for the value of E_f, the loss by friction, have been given by various writers, and an excellent discussion of the results obtained by use of different formulæ is given in *Engineering*, March 19, 1897, by Mr. Arthur J. Martin.

It seemed desirable, after obtaining unsatisfactory results with other formulæ, to reduce all experimental results by the formula given by Unwin in article "Hydro-Mechanics," page 484 (*Encyclopædia Britannica*), which is reduced from Weisbach. In accordance with this formula the frictional work is for the length of pipe L.

$$E_f = f \frac{V^2}{2g} \frac{4}{d} WL = f \frac{V^2}{g} \frac{2}{d} WL \quad . \quad . \quad . \quad . \quad (5)$$

In which f = coefficient of friction, V = velocity in feet per second, d = diameter in feet, W = weight of fluid per second. This substituted for E_f in equation (4) gives

$$E - E_c = f \frac{V^2}{g} \frac{2}{d} WL \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The value of E_c was found by experiment to average 1,700 foot-pounds per second for the pipe, protected as well as possible by covering.

The value of f is dependent upon the velocity of the fluid in the pipe, and is given as follows by

Eytelwein as $f = a + \frac{b}{v} = .0056 + \frac{.00144}{v}$.

Weisbach, $f = a + \frac{b}{\sqrt{v}} = .0036 + \frac{.0043}{\sqrt{v}}$.

Arson, $f = a + \frac{b}{v} = .006 \text{ to } .009 + \frac{.009 \text{ to } .04}{v}$,

in which a and b are constants.

The value of f, the coefficient of friction, is also stated to vary with the diameter. Unwin found that for velocities of one hundred feet per second it could be expressed by the formula:

$$f = K\left(1 + \frac{3}{10d}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

in which K is a coefficient to be found by experiment, and d is the diameter of the pipe in feet. The values of a and b found in Arson's experiments, indicates that the coefficient K would probably vary between one and three per cent. for velocities from between fifty and one hundred and fifty feet per second.* This being a small amount, it was considered sufficiently accurate to use the expression in formula (7) for reducing all the experimental results.

Formula (6) may be reduced to the form

$$E - E_c = Wh = f\frac{V^2}{g}\frac{2}{d}WL \quad . \quad . \quad . \quad . \quad (8)$$

from which the loss of head

$$h = f\frac{V^2}{g}\frac{2}{d}L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Let p equal loss of pressure in pounds per square inch, and D density of the fluid or weight per cubic foot, then will

$$p = \frac{h\,D}{144};$$

* See "Hydro-Mechanics," vol. xii., *Encyclopædia Britannica*.

substituting this value in (9)

$$h = \frac{144p}{D} = f\frac{V^2}{g}\frac{2}{d}L = 2\frac{V^2}{gd}LK\left(1 + \frac{3}{10d}\right);$$

from this by transposition

$$p = \frac{DV^2LK}{72gd}\left(1 + \frac{3}{10d}\right) \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Let w' equal flow in pounds of steam per minute, d' equal diameter of the pipe in inches, then we shall have

$$V = \frac{w}{60} \cdot \frac{1}{D} \cdot \frac{4}{\pi d^2} = \frac{w'}{15\pi Dd^2} = \frac{9.6w'}{\pi Dd'^2};$$

substituting in (10) we have

$$p = \frac{(9.6)^2 DLKw'^2}{6gd'^5D^2\pi^2}\left(1 + \frac{3.6}{d'}\right) = \frac{1}{20.663}K\left(1 + \frac{3.6}{d'}\right)\frac{w'^2L}{Dd'^5} \quad (11)$$

For a two-inch pipe the value of p will reduce to the following:

$$p = 1{,}447.2\frac{w'^2K}{D};$$

from this by transposition

$$K = \frac{pD}{1{,}447.2w'^2}.$$

Description of Experiment.—In Fig. 116 is shown the general plan of the experimental pipe line, arranged for measuring the friction in both the pipe and fittings. The steam enters at J, and passes through the separator S_1; the initial quality and pressure of steam are obtained by the throttling calorimeter C_1 and the gauge G_1. The pressure and quality are again measured at G_2 and C_2 just before entering the separator S_2, at the end of the pipe line. The length of pipe from G_1 to G_2 was 90 feet. The pressure and quality of steam were observed at C_3 and G_3 just before entering the fittings. The final pressure is measured by G_4 after the steam has passed through the fittings and is entering the orifice O. It is thus seen that the pressure and quality of the steam at each important stage may be observed, and the

velocity and the weight of steam delivered at the orifice O, computed.

RESULTS OF THE TEST.

The principal data and results of the tests are given in tabular form. The log sheets are quite voluminous, and are not given.

The following table gives the reduced data and results of experiments with covered pipe 2 inches in diameter, 90 feet in length, and with steam discharging through a $\frac{3}{4}$-inch nozzle:

No.	Pressures. Initial. P_1.	Final. P_2.	Drop. P.	Average. $\frac{1}{2}(P_1 + P_2)$.	Density. D	Weight per Second. W.	Velocity in Feet per Second. V.	Value of Coefficient. K.
1.....	106.77	101.95	4.82	104.3	.23969	.6231	122	.00276
2.....	104.37	99 3	5.01	101.8	.23420	.6073	119	.00222
3.....	109.1	104.7	4.4	106 9	.24525	.6436	120	.00180
4	107 5	102.6	4.9	105.0	.24113	.6259	118	.00205

The following table gives the reduced data and results of experiments with covered pipe 2 inches in diameter, 90 feet in length, and with steam discharging through a $\frac{7}{8}$-inch nozzle:

No.	Pressures. Initial. P_1.	Final. P_2.	Drop. P.	Average. $\frac{1}{2}(P_1 + P_2)$.	Density. D.	Weight per Second. W.	Velocity in Feet per Second. V.	Value of Coefficient. K.
1.....	101.6	92.8	8.8	97.2	.2242	.7580	156	.00237
2.....	108.87	100.95	7.92	104.91	.2320	.8278	162	.00221
3.....	108 87	101	8.5	105.25	.2411	.7574	144	.00246
4.....	99.55	92	7.5	95 37	.2194	.7576	164	.00198
5.....	98.75	91.45	7 3	95 10	.2193	.7387	155	.00202
6.....	103.5	96.2	7.3	99.8	.2300	.7860	156	.00241

For the first series of experiments the highest value of K is .00276; the lowest, .00180; the average, .00221. The lowest value is considerably less than the others, and may have been due to some error of observation; neglecting this, the average value of $K = .00234$.

The highest value of K in the second series of observations is .00237; lowest, .00198; the average, .00224. The average, neglecting the highest and lowest values, is .00228.

A series of tests was made with the pipe uncovered, allowing

the steam to discharge through $\frac{5}{8}$, $\frac{3}{4}$, and $\frac{7}{8}$-inch nozzles. The results of this test were irregular and not satisfactory, probably because of the fact that the condensation was irregular, and large errors were caused in correcting for it. The general results, as compared with the tests of the covered pipe, indicate smaller values, doubtless due to an over-correction for condensation.

The following are the results obtained in the tests of the uncovered pipe:

Discharge Nozzle.	½-inch.	¾-inch.	⅞-inch.
	Values of K.	K.	K.
April 30, 1897	.00197	.00290	.00188
April 22, 1897	.....	.00195	.00093*
April 21, 1897	.00163		
May 11, 1897	.....	.00174	.00189
May 19, 1897	.....	.0036*	.0027
Average	.0018	.00254	.0017
Average neglecting values marked with *	.....	.....	.00208

In these experiments the highest value of K was .00360; the lowest, .00093. The average, neglecting the extreme value, is .00298.

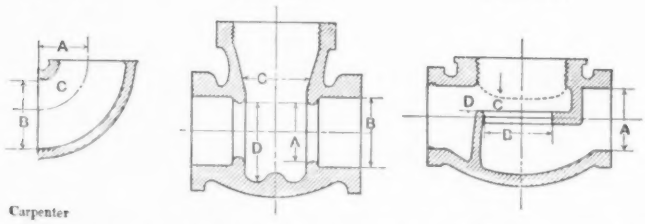
The experiments cited indicate the following value of K: First set, .00234; second set, .00228; third set, .00208; average, .00223.

During the fall of 1897 an exceptional opportunity was afforded by a boiler test at Oriskany, N. Y., of determining the coefficient of friction of steam. Three Sterling boilers of 175 horse-power each discharged the total evaporation through a 3-inch pipe 95 feet long, provided at its extremity with a nozzle 2.32 inches in diameter during a boiler test extending over 12 hours of time. The measurement of the water supplied the boiler during the test checks with the calculation of the steam discharged by the nozzle within less than 1 per cent. The value of K, as found from this test, was .0026. The data of the test were as follows: The gauge pressure at initial end of the pipe, 66 pounds; at discharge orifice, 41 pounds; drop in pressure, 25 pounds; mean absolute pressure, 78.2; velocity, 960 feet. Steam entering pipe was throttled from 100 pounds and slightly superheated.

Flow of Steam in Elbows and Valves.

Dimensions 90-*Degree Elbows.*—We will first call attention to a few measurements made on elbows, globe and gate valves. The following table for elbows shows that the makers have

FIG. 117. FIG. 118. FIG. 119.



Carpenter

made the mean radius *A* (Fig. 117) of the elbow, approximately equal to the threaded opening *B*. That is, the turn is made as short as possible, thus producing a most compact elbow.

NINETY-DEGREE ELBOWS.

Nominal Size.	Actual Dimensions.			
	A	B	C	$\frac{\frac{1}{2}B}{A}$
2	$2\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	.50
$2\frac{1}{2}$	$2\frac{5}{8}$	$2\frac{5}{8}$	$2\frac{7}{8}$	.50
3	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{2}$	.50
4	$4\frac{1}{4}$	$4\frac{1}{4}$	$4\frac{3}{8}$	.50
$4\frac{1}{2}$	4	$4\frac{3}{4}$	5	.59
6	6	$6\frac{1}{2}$	$6\frac{7}{8}$	.54

Dimensions Globe Valves.—The general form of the valve shell is shown in Fig. 119, and the dimensions of a few valves are given in the following tables. It was found that the maximum opening *C* (see Fig. 119) in the clear for a bevelled seat valve is not so great as for a disk valve. The dotted line represents the valve face when at its maximum opening.

BEVELLED SEAT GLOBE VALVES.

Nominal Size.	Approximate Dimensions.			
	A	B	C	D
$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$
2	$2\frac{1}{4}$	2	$\frac{13}{16}$	$1\frac{3}{16}$
$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{4}$	1

It is a matter of interest to note that the opening B and the nominal size are the same in inches.

A few dimensions for the disk-seat Jenkins valve are given in the next table, the shell of which is similar to that of Fig. 119.

JENKINS DISK-SEAT GLOBE VALVE.

Nominal Size.	Approximate Dimensions.			
	A	B	C	D
$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
2	$2\frac{1}{4}$	2	$1\frac{3}{16}$	$\frac{13}{16}$

Gate Valves.—It will be noted by the following table for gate valves that the opening A (see Fig. 118) is equal to the nominal size of the valve. These nominal sizes, it must be understood, refer to nominal sizes of pipes for which the fittings are adapted.

GATE VALVES. KENNEDY TYPE.

Nominal Size.	Approximate Dimensions.			
	A	B	C	D
1	1	$1\frac{1}{4}$	$\frac{7}{8}$	$1\frac{1}{4}$
$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{3}{4}$	1	2
2	2	$2\frac{1}{4}$	$1\frac{3}{8}$	$2\frac{3}{8}$
$2\frac{1}{2}$	2	$2\frac{3}{4}$	$2\frac{1}{4}$	$3\frac{3}{4}$

Method of Testing Friction in Elbows and Valves.—The method of testing involved simply the measurement of drop of pressure. The two important stages for present consideration are, first, the flow of the steam from G_1 to G_2 (Fig. 116); and, second, the flow from G_3 to G_4. The first stage involves—for known conditions of flow—the drop of steam pressure in a pipe line of known length; the second stage involves the drop in the elbows or valves placed in sequence. In Fig. 116 the elbows $E\,E$ are shown in position; the dark lines represent the short nipples used to connect the fittings. The nipples were so short that the drop of pressure in them was negligible. The velocity of the steam was varied from fifty to one hundred and twenty-five feet per second by different diameters of orifice O. As many as sixteen valves or elbows were placed in series, and the various drops in

pressure were measured for the different velocities; then the number of valves or elbows was decreased, and the measurement of drop repeated. From the data so found the drop in one valve or elbow for different velocities was computed. The globe valves were of the slightly bevelled type; the gate valves were of the Kennedy pattern, and the elbows were confined to the ninety-degree type.

Simultaneously with the readings for the valves and elbows, readings were taken for the pipe line, and thus comparison was made directly. The weight through the fittings is less than the weight through the line, by the amount taken out by the separator S_2, and the average density was also less.

The formulæ adopted for purposes of comparison are those given by Weisbach (1), as follows:

For loss of head H_p in feet for pipes,

$$H_p = f_p \frac{4L}{d} \frac{V_2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and for loss of H_e in feet for elbows,

$$H_e = f_e \frac{V_2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

We will assume that (2) applies also to valves, and from the data we find:

$$f_p = K\left(1 + \frac{3}{10d}\right)$$

f_e (for 90° elbows) $= 5.6.$
f_g (for globe valves) $= 8.2.$
f_v (for gate valves) $= .005.$

This indicates that the drop through a gate valve is practically negligible for most cases.

Equating (1) and (2), an expression for the equivalent length of one elbow or one valve in feet of pipe of the same diameter is obtained.

$$L' = \frac{f_e}{K}\frac{d'}{4} - \left(1 + \frac{3.6}{d'}\right) = \frac{f\,d'}{4K\left(1 + \frac{3.6}{d'}\right)}$$

where d' and L' are both in inches.

By substitution of the proper values of K and f_e, it is found that the length of pipe in inches equivalent to one elbow is

$$L' = 520d' - \left(1 + \frac{3.6}{d'}\right)$$

and to one globe valve

$$L' = 706d' - \left(1 + \frac{3.6}{d'}\right)$$

The values given in "Steam," Babcock & Wilcox, page 89, by the late George H. Babcock, are as follows: for one globe valve

$$L' = 114d' - \left(1 + \frac{3.6}{d'}\right)$$

for one elbow $L' = \frac{2}{3}$ that of a globe valve, or

$$L' = 76d' - \left(1 + \frac{3.6}{d'}\right)$$

It will be seen that the experiments give values which are about seven times as great as those given in "Steam."

An opportunity for checking the results of the test quoted on flow of steam in pipes was afforded by an experiment made by Green and Stratford for the purpose of obtaining drops in pressure through certain steam-pipe line with steam flowing at different velocities, and is described in the thesis which they presented for graduation in 1895.* In this case there were 25 feet of 8-inch pipe leading from boiler room to engine room thence 192 feet of 3-inch covered pipe leading from the engine-room to the mechanical laboratory of Sibley College; thence 112 feet of 2½-inch covered pipe leading to the new Sibley building; thence 115 feet of 2-inch bare pipe, at the end of which there was the orifice of discharge. There were twelve 90-degree elbows in the 2½-inch pipe, and the same in the 2-inch pipe, for which allowance is made in accordance with the friction found for the test described. This allowance amounts to 80 feet of 2½-inch pipe, and 60 feet of 2-inch pipe. The following table gives the data from Green and Stratford's experiments—the calculated drop of pressure in the 2-inch pipe and

* See Library of Cornell University.

that value of the coefficient K required to give results agreeing with the experiment.

The average value of the coefficient for these experiments is $K = .00220$.

No.*	PRESSURES.			Weight Discharged per Second. W.	Velocity Feet per Second. V.	Drop in 2-inch Pipe. Lbs.	Density Weight per Cubic Foot. D.	K.
	Mechan. Labor.	At Orifice.	Drop in Pressure.					
1	42.57	42.10	0.47	.1179	53	.36	.1018	.00374
2	53.55	52.70	0.85	.1475	54	.65	.1267	.00142
5	82.20	81.00	1.20	.2268	57	.94	.1900	.00130
6	91.86	89.90	1.96	.2517	57	1.52	.2085	.00186
7	103.64	101.20	2.44	.2833	57	1.88	.2346	.00206
8	111.00	109.10	2.00	.3000	57	1.54	.2520	.00112
9	121.70	117.70	4.00	.3295	58	3.07	.2713	.00288
10	131.30	127.20	4.10	.3560	58	3.15	.2930	
11	87.79	44.91	41.88	.8000	345	33.50		
12	86.87	55.35	31.52	.7514	267	25.21		
13	88.74	76.91	11.83	.5050	132	9.06	Average.	.00220

* Numbered as in thesis of Green and Stratford.

The values of the coefficient obtained in these experiments do not differ materially from those obtained by the experiments cited in vol. xii., *Encyclopædia Britannica*, article "Hydro-Mechanics." Professor Unwin gives formulæ for the coefficient of friction for flow of *water* through pipes, as follows: $f = .005\ (1 + \frac{1}{12d})$, and for a slightly incrusted pipe, $f = .01\ (1 + \frac{1}{12d})$. For the flow of *air*, by experiments at St. Gothard tunnel:

$$f = .0028\left(1 + \frac{3}{10d}\right) \text{ for air.}$$

By experiments by M. Arson on pipes which were probably rougher,

$$f = 0.005\left(1 + \frac{3}{10d}\right) \text{ for air.}$$

The experiments at Sibley College, which have been cited, indicate a value of the coefficient of friction for steam flowing in pipes, as follows:

$$f = .00223\left(1 + \frac{3}{10d}\right) \text{ for steam.}$$

The experiments at Oriskany indicated the coefficient of friction:

$$f = .0026\left(1 + \frac{3}{10d}\right) \text{ for steam.}$$

The tables were calculated for a coefficient of friction,

$$f = .0027\left(1 + \frac{3}{10d}\right) \text{ for steam.}$$

In the last five formulæ, d, the diameter, is to be taken in feet.

The complete expression for loss of head in pounds, p, is obtained by substituting the value 0.0127 for K in equation (11), in which case

$$p = 0.000131\left(1 + \frac{3.6}{d'}\right)\frac{w'^2 L}{Dd'^5},$$

in which d' = diameter in inches, L = length in feet, D = density.

Explanation of Tables.

The first table given is based upon formula 11, and is arranged in the form suggested by Martin in his paper on "The Flow of Gases," previously referred to. The table is computed by taking the value of K equal to .0027, which, although 15 per cent. higher than the average obtained in the experiments cited, seems to have been about the probable value in at least one case examined. For practical conditions it is rather better to have an allowance in the pipes, for an excess of friction, than to have the reverse condition true. The results in the table are calculated for pipes 1,000 feet long, and having a coefficient of friction as given above.

The left-hand half of Table I. gives the discharge in pounds per minute for pipes of various diameters corresponding to drop of pressure as given on the right-hand side in the same horizontal line; *e.g.*, a 24-inch pipe discharges 14,000 pounds per minute, a 22-inch pipe under exactly similar conditions will discharge 11,188 pounds, and a 6 inch pipe 370 pounds for a drop in pressure as in same horizontal line on right-hand side, for instance, with a drop of 16.4 pounds at 101 pounds pressure; so the discharges may be found for any other given diameter by following the same horizontal line.

In the right-hand body of the table are given the drops in pressure corresponding to the discharges in the same horizontal line. These drops of pressure are computed for a constant length of 1,000 feet, and for the densities and corresponding absolute pressures as given in first two lines above.

In using the table the absolute pressures should be taken as the mean of the initial and final pressures in computing the carrying capacity; *e.g.*, if the initial pressure of the steam entering the pipe is 108.2 pounds absolute, and the final pressure upon leaving the pipe is 91.8 pounds, the mean absolute pressure is 100 pounds, and the total loss in pressure is 16.4 pounds.

It may be determined from the table, for instance, that a 6-inch pipe, 1,000 feet in length, with a drop of pressure of 16.4 pounds, and average absolute pressure of 100 pounds, will deliver 371 pounds per minute; and similarly a 7-inch pipe will deliver 560 pounds; a 22-inch pipe 11,188 pounds, and a 24-inch pipe 14,000 pounds, and similarly for any other diameter in the same horizontal line.

If the discharge for any diameter of pipe be taken from the left-hand side of the table, its corresponding drop for any mean absolute pressure may be found in the right-hand side of the table in the same horizontal line. From formula 11 for a given flow of steam and diameter of pipe the drop in pressure is proportional to the length, and if discharge values for other lengths of pipe are required they may be obtained by proportion. For example, to find the weight discharged per minute in a 6-inch pipe 200 feet in length, and having an absolute mean pressure in pipe of 125 pounds, and a drop in pressure of 2 pounds. For this case the tabular flow would correspond to a drop five times that given, or 10 pounds. Looking under steam pressure of 125 pounds, for drop of 10 pounds, we find it is to be interpolated between 9.74 and 11.40. The corresponding discharge in the same horizontal line under 6 inches is 318 + 10 = 328 pounds. For a length of 120 feet, with gauge pressure 90 and mean absolute pressure 104, drop 1 pound, we should have drop to correspond with table, $\frac{1000}{120} = 8.33$ pounds. Looking under 104 pounds pressure for 8.33, we find it must be interpolated * between 8.05 and 9.66. The discharge is $265 + \frac{28}{161} \cdot 27 = 265 + 4.7 = 270$, nearly.

* This interpolation, though sufficiently accurate for practical purposes, is not exact, as the discharge varies with the square root of the drop in pressure. The discharge should be greater for this case.

In the table given in "Steam," by Babcock & Wilcox, the discharge for this condition is given as 281 pounds per minute.

Tables II. *and* III.—These tables, it is thought, will prove more convenient for use than Table I.; they are based upon the same

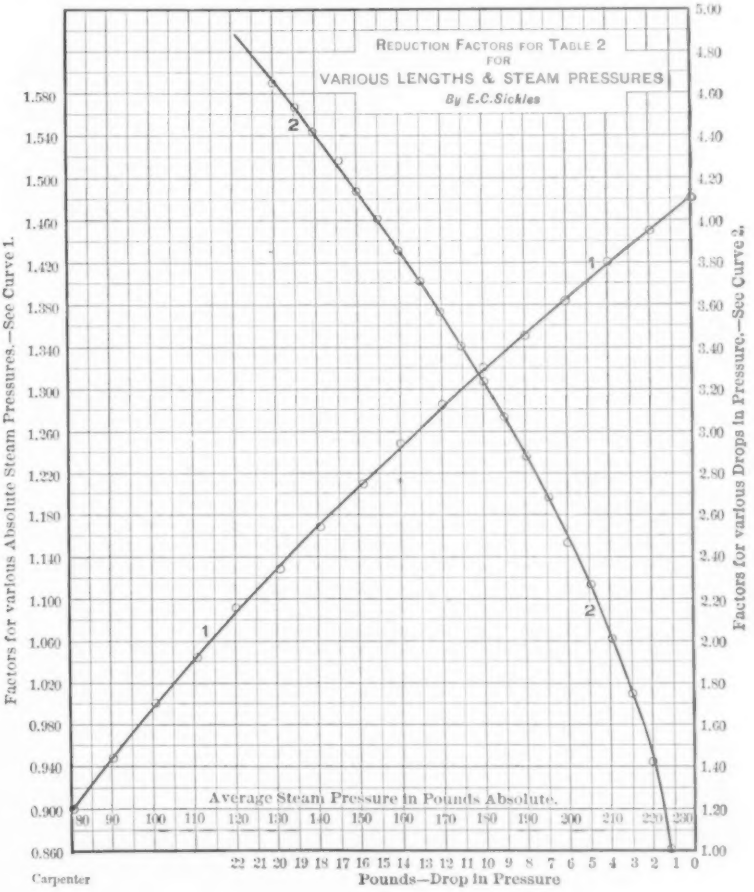


Fig. 120.

formula, and are deduced by making every factor constant except the diameter, length of pipe, and discharge. The left hand vertical column of the table contains the diameters (d) of the pipes, and the top horizontal column the length (L) in feet, while the body of the table gives values of (W) the pounds discharged per minute.

TABLE II.—FLOW OF STEAM IN PIPES.

DELIVERY IN POUNDS PER MINUTE BY PIPES OF GIVEN DIAMETERS AND LENGTHS.

(ABSOLUTE PRESSURE ONE HUNDRED POUNDS—DROP IN PRESSURE ONE POUND.)

CALCULATED BY E. C. SICKLES.

Diam. in Inches.	LENGTH IN FEET.																							
	25	50	75	100	125	150	175	200	225	250	275	300	325	350	375	400	425	450	475	500	525	550	575	600
¾	2.20	1.55	1.27	1.10	.983	.898	.831	.700	.733	.698	.659	.633	.611	.589	.568	.550	.532	.518	.506	.491	.481	.468	.457	.450
1	4.40	3.10	2.54	2.20	1.96	1.79	1.66	1.55	1.46	1.39	1.32	1.26	1.22	1.17	1.13	1.10	1.06	1.03	1.00	.982	.962	.932	.914	.900
1¼	9 76	6.90	5.63	4.88	4.36	3.93	3.69	3.44	3.26	3.08	2.92	2.81	2.72	2.62	2.52	2.44	2.36	2 29	2.24	2.18	2.13	2.07	2.03	1.96
1½	13.04	9.29	7.64	6.52	5.83	5.83	4.95	4.62	4.35	4.14	3.92	3.77	3.62	3.50	3.36	3.26	3.16	3.07	3.00	2.91	2.85	2.77	2.72	2.67
2	30.54	21.6	17.6	15.2	13.6	12.50	11.58	10.8	10.2	9.82	9.32	8.82	8.40	8.37	7.87	7.63	7.40	7.18	7.01	6.81	6.68	6.51	6.36	6.23
2½	50.8	35.9	29.3	25.4	22.7	20.8	19.2	17.9	16.9	16.2	15.2	14.62	14.4	13.6	13.1	12.7	12.3	11.9	11.6	11.3	11.0	10 7	10.5	10.4

Diam. in Inches.	LENGTH IN FEET.																							
	50	100	175	250	325	400	475	550	625	700	775	850	925	1,000	1,075	1,150	1,225	1,300	1,375	1,450	1,525	1,600	1,675	1,750
3	65.17	46.0	35.0	29.2	25.6	23.0	21.2	19.6	18.4	17.4	16.5	15.8	15.2	14.5	14.0	13.6	13.1	12.7	12.4	12.10	11.82	11.5	11.2	11.0
3½	98.3	69.5	52.7	44.6	38.6	34.7	31.9	29.6	27.7	26.3	24.92	23.8	22.8	22.0	21.3	20.5	19.8	19.2	18.6	18.2	17.8	17.4	16.9	16.6
4	138.1	97.6	73.8	61.8	54.2	48.8	44.8	41.5	39.1	36.7	35.0	33.5	32.1	30.8	29.8	28.7	27.8	27.1	26.3	25.6	25.0	24.4	23.8	23.3
4½	187.9	132.9	102.0	84.1	73.8	66.45	61.2	56.6	53.1	50.1	47.7	45.6	43.7	42.0	40.51	39.2	38.0	36.75	35.75	34.75	34.0	33.2	32.4	31.8
5	255.6	180.7	136.2	114.3	102.2	90.5	83.3	76.9	72.1	68.2	65.1	62.1	59.6	57.1	55.1	53.2	51.6	50.0	48.6	47.2	46.3	45.2	44.2	43 2
6	419.4	296.5	224.1	187.4	164.2	148.2	136.1	125.8	118.2	111.8	106.2	101.8	97.3	93.7	90.2	87.2	84.2	82.2	79.7	77.6	75.7	74.1	72.2	71.0

TABLE III.—FLOW OF STEAM IN PIPES.

Delivery in Pounds per Minute by Pipes of Given Diameters and Lengths.

(Absolute Pressure One Hundred Pounds—Drop in Pressure One Pound.)

Calculated by E. C. Sickles.

Diam. in Inches.	Length in Feet.																							
	50	100	175	250	325	400	475	550	625	700	775	850	925	1,000	1,075	1,150	1,225	1,300	1,375	1,450	1,525	1,600	1,675	1,750
7	618	437	330	276	242	218	200	186	174	165	157	150	143	138	133	129	125	121	118	115	112	109	107	104
8	890	624	472	394	346	312	286	266	249	236	224	214	205	197	190	184	178	173	168	164	160	156	152	149
9	1,206	853	644	539	473	426	391	363	341	322	306	292	280	269	260	251	243	236	230	224	218	213	208	204
10	1,592	1,126	851	712	624	563	516	480	450	425	404	386	370	356	343	332	321	312	303	295	288	281	275	269
11	2,046	1,447	1,093	915	803	723	664	617	574	546	519	496	476	457	441	426	413	401	390	380	370	361	353	346
12	2,575	1,887	1,428	1,192	1,044	943	852	803	752	714	679	648	620	598	575	557	539	523	509	494	487	472	461	451
13	3,165	2,298	1,692	1,415	1,242	1,119	1,027	954	891	846	804	767	736	707	682	660	639	620	603	588	573	559	547	535

Diam. in Inches.	Length in Feet.																							
	50	100	200	300	400	500	600	700	800	900	1,000	1,100	1,200	1,300	1,400	1,500	1,600	1,700	1,800	1,900	2,000	2,100	2,200	2,300
14	3,838	2,714	1,920	1,567	1,357	1,213	1,108	1,025	959	904	858	818	783	752	725	700	678	658	639	622	606	592	578	566
15	4,596	3,250	2,300	1,873	1,625	1,453	1,327	1,228	1,149	1,083	1,028	980	938	901	868	839	812	788	766	745	726	709	693	678
16	5,630	4,000	2,830	2,315	2,000	1,785	1,638	1,513	1,413	1,333	1,268	1,208	1,153	1,108	1,072	1,034	1,000	972	945	916	895	874	854	834
17	6,375	4,500	3,210	2,635	2,250	2,021	1,841	1,702	1,591	1,500	1,424	1,357	1,302	1,248	1,200	1,162	1,124	1,087	1,061	1,033	1,006	982	962	940
18	7,370	5,211	3,685	3,008	2,605	2,330	2,128	1,970	1,843	1,737	1,648	1,571	1,504	1,445	1,393	1,345	1,302	1,264	1,228	1,192	1,165	1,137	1,111	1,087
19	8,472	5,992	4,237	3,459	2,996	2,680	2,447	2,265	2,119	1,997	1,895	1,807	1,730	1,662	1,602	1,547	1,498	1,453	1,412	1,374	1,340	1,307	1,278	1,249
20	9,240	6,839	4,835	3,948	3,419	3,059	2,793	2,585	2,418	2,279	2,163	2,062	1,974	1,897	1,828	1,766	1,709	1,658	1,612	1,569	1,529	1,492	1,458	1,426
22	12,365	8,743	6,183	5,048	4,371	3,910	3,570	3,278	3,093	2,914	2,765	2,637	2,524	2,406	2,337	2,257	2,185	2,121	2,061	2,005	1,955	1,908	1,864	1,823
24	15,943	11,308	7,990	6,535	5,650	5,065	4,522	4,435	3,995	3,769	3,580	3,418	3,271	3,141	3,023	2,921	2,820	2,750	2,665	2,585	2,535	2,472	2,415	2,361

Thus, for instance, a 10-inch pipe 50 feet long will deliver 1,592 pounds of steam per minute with a drop of one pound in pressure, if there exists an average absolute pressure of 100 pounds; or, if all other conditions hold except the length of pipe, which varies, it may be seen that for 100 feet the discharge is 1,126 pounds, for 175 feet 851 pounds, and so for any number of feet given in the table.

If any intermediate length of pipe is used other than those given in the tables, the discharges given by the tables may be corrected by consideration of the fact that the weight of discharge is inversely proportional to the square root of the length of the pipe.

To meet the conditions where other average absolute pressures than 100 pounds exist, and higher drops than one pound are assumed, it is only necessary to use suitable factors which are calculated by means of the fundamental formula, and graphically represented by Curves 1 and 2.

As an illustration of the use of the tables and curves, suppose it is desired to find what size pipe will be required to deliver 1,000 pounds of steam per minute a distance of 1,000 feet, the initial pressure of the steam being 157.5 pounds, and the final 152.5 pounds by gauge. Solution.—It will be best to reduce all conditions to those of the tables and find the discharge, and from this the size of the required pipe. Looking at Curve 2, we find the factor of discharge for a 5-pound drop is about 2.23 times that for a 1-pound drop. Therefore, dividing the required discharge of 1,000 pounds by 2.23, we have about 450 pounds discharge for a 1-pound drop.

Again, the average pressure is 155 + 15, or 170 pounds absolute, and from Curve 1 it may be found the factor of discharge is 1.248 greater than for 100 pounds absolute. Therefore, dividing 450 pounds by 1.284 we have 350 pounds on the basis of the conditions given by the tables; and looking under 1,000 feet lengths for the discharge nearest to 350 pounds, we find a 10-inch pipe discharges 356 pounds per minute; therefore, it would be satisfactory.

DISCUSSION.

Mr. Wm. Kent.—The author gives on page 346 the well-known formula for drop of pressure :

$$h = f \frac{V^2}{g} \frac{2}{d} L,$$

f being supposed to be the coefficient of friction. Whether that coefficient of friction is variable or constant is a question.

On top of page 346 he gives Eytelwein's, Weisbach's, and Arson's ideas, saying that the coefficient of friction depends on the velocity of the fluid ; then a little lower down that Unwin finds it varies with the diameter, and for one velocity only he finds that it equals

$$K\left(1 + \frac{3}{10d}\right).$$

So we may say the coefficient of friction, according to various authorities, varies according to laws upon which they do not agree. It may vary as some constant a and some other constant b divided by the velocity or by the square root of the velocity. So we see that the value of f is an exceedingly uncertain sort of thing. All old authorities—Weisbach, Arson, Eytelwein, Unwin, and others—have reasoned from a great variety of data, and Professor Carpenter adopted Unwin's results without investigation, as Eytelwein, Weisbach, and Arson said their discrepancies were or are due to errors of observation, but assumes that K should be a constant, and takes an average of the several values ; and then, after some other assumptions, he arrives at the value of f by using Unwin's formula, and putting instead of K a constant value .0027. He then constructs this very long table, and asks us to accept it as a good table for the flow of steam.

Professor Jacobus.—Some extensive investigations relating to the flow of steam in pipes was made by M. Ledoux, and published in *Annales Des Mines*, vol. ii., 1892. A translation was printed in the *Stevens' Indicator*. I should like the privilege of

comparing the tables given by Professor Carpenter with the results of the Ledoux formula, and of adding the same to the discussion.

*Professor Carpenter.**—At the request of Professor Jacobus made by letter since the annual meeting the writer has carefully examined the paper by M. Ledoux to which he refers, and submits a comparison of the formula deduced by Ledoux and also gives a synopsis of the important data.

The general formula for flow of steam adopted by Ledoux is essentially of the same nature as that stated in the paper, except that Ledoux assumes the coefficient of friction, f, as constant for all diameters of pipes and velocities of the flowing steam. He gives in the same paper the results of a series of experiments on the flow of compressed air in pipes, and for this case he also uses a formula of the same general nature. The method of performing experiments was similar to that described in the paper, inasmuch as he determined the amount of steam supplied, the amount discharged, and the condensation and loss of pressure, and from these data computed the coefficient of friction.

Ledoux's experiments were made on a larger scale than those described in the paper, and, consequently, his results are of great value. The experiments, referred to in the paper by the author, were made on pipes respectively 1-inch, 1¼-inch, 2-inch, and 3-inch, and with lengths varying from 90 to 250 feet. Those made by Ledoux were on pipes from 47 millimetres (1.85 inches), 71 millimetres (2.79 inches), 75 millimetres (2.95 inches), and 100 millimetres (3.94 inches) in diameter, and with lengths varying from about 100 metres (328 feet) to 330 metres (1,082 feet). He also quotes an experiment made by Gutermuth on a pipe 140 millimetres (5.51 inches) in diameter. He made an extensive series of observations on the amount of condensation, and corrected for the same in his results, although he concludes that the correction for condensation is less than the unavoidable errors of observation. It will be noticed that the coefficients of friction deduced from his observations vary within wide limits, and that the errors of observations must have been, in many cases, of considerable magnitude.

Instead of using the value of the density of steam or its weight

* Author's closure under the Rules.

per unit of volume, as in the formulæ given by the writer, he takes for the value of the density of steam

$D = 0.606\ p^{0.94}$ when p is expressed in atmospheres.

$D = 0.588\ p^{0.94}$ when p is in kilograms per square centimetre.

By referring to a steam table it will be found that

$D = 0.003\ p^{0.94}$ when p is expressed in pounds per square inch.

Thus when

$p = 40$; $p^{.94} = 0.00292$. $\quad$ $p = 100$; $p^{.94} = 0.00294$.

$p = 80$; $p^{.94} = 0.00293$. $\quad$ $p = 150$; $p^{.94} = 0.0033$.

Ledoux gives as a general value for the coefficient of friction

$$f = \frac{(p_0^{1.94} - p_1^{1.94})d^5}{0.000535Lm_0(m_0 - 0.000436jdL)}, \quad . \quad . \quad (14_L)$$

in which p = pressures in atmospheres, initials p, final p^1; d = diameter in metres, L, length in metres; m_0 = weight per second supplied in kilograms; j = weight of condensation per square metre per hour, so that $0.000436\ jdL$ is a correction for condensation. Allowing a numerical value for condensation, Ledoux deduces the following value for the diameter:

$$d = \sqrt[5]{\frac{0.000502fm_0^2L}{p_0^{1.94} - p_1^{1.94}}}.$$

The mean value of f from his experiments was 0.0011, which substituted in the above gives the following values:

$$d = \sqrt[5]{\frac{0.000000552m_0^2L}{p_0^{1.94} - p_1^{1.94}}};$$

$$d = 0.056\sqrt[5]{\frac{m_0^2L}{p_0^{1.94} - p_1^{1.94}}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15_L)$$

the latter being the one which Ledoux gives as the most probable value of the diameter from his experiments, and is the one which he advises to be used for all practical purposes.

As an example, suppose that in a pipe 300 metres in length 1 kilogram per second is supplied when initial pressure is 10

atmospheres, and final pressure 9.5 atmospheres, to find the diameter of the pipe. Substituting these values in (15_L), we have

$$d = 0.056 \sqrt[5]{\frac{300}{10^{1.94} - 9.5^{1.94}}} = 0.56 \sqrt[5]{\frac{300}{87.1 - 78.8}}$$
$$= 0.115 \text{ metre (4.39 inches)}.$$

In a similar manner for a flow of 4 kilograms per second d equals 0.2 metre (7.87 inches), and for a flow of 10 kilograms per second (1,320 pounds per minute) the diameter equals 0.288 metre (11.33 inches).

Taking the results from the table given by the author for the same conditions, we have initial pressure 10 atmospheres equals 147 pounds, final pressure 9.5 atmospheres equals 139.65 pounds, mean pressure 144.4 pounds, drop in pressure 7.35 pounds. The flow of 1 kilogram per second equals 132 pounds per minute, of 4 kilograms per second equals 528 pounds per minute; 10 kilograms per second equals 1,320 pounds per minute.

Looking at right-hand side of table under 145 pounds pressure for drop nearest to 7.35 pounds, we note 7.31 pounds in fourth horizontal line from top. From this point follow to the left in the same horizontal line for the nearest number corresponding to the given flow of steam, which in the first instance is 132 pounds. The delivery given by the table for a 4-inch pipe under these conditions is 97 pounds, and for a 5-inch pipe 178 pounds By interpolation on the supposition that the variation of flow is proportional to the square root of the fifth power of the diameter, a pipe slightly under 4.5 inches will deliver 132 pounds per minute. For delivery of 528 pounds per minute, a pipe will be required larger than 7 inches in diameter and smaller than 8 inches. For delivery of 1,320 pounds a pipe will be required about 10.7 inches in diameter, or about one-half inch smaller than required by the formula of Ledoux. The substitution of the numerical values in the formula (17) given in the paper, for a flow of 132 pounds per minute gives a diameter of 4.4 inches, which is almost exactly the same as that obtained from the Ledoux formula.

The examination would indicate that although there is a slight difference in the method of treatment of this problem by Ledoux and myself there is a very close agreement for pipe sizes with diameters between 3 and 8 inches; that for smaller sizes of pipes

the writer's table requires larger diameters for a given flow and for larger sizes a smaller diameter for a given flow, for the reason that the writer assumed as agreeing with his experiments that the coefficient of friction diminished with the increase of the diameter, while Ledoux considers it constant. Considering only the commercial sizes of pipes, the table in the paper gives results which are in practical agreement with the formula of Ledoux.

A comparison of the formulæ given in the paper with those of Ledoux may be made as follows. Thus in equation (11) of the paper substituting for $K\left(1+\frac{3.6}{d'}\right)$ the symbol of the coefficient of friction f, we shall have by transposing

$$f=\frac{20.663\,pDd'^5}{w'^2L} \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

In the above formula p = drop in pressure = $p_0 - p_1$, and D = mean density; so that, numerically, pD is very nearly equal to $p_0D_0 - p_1D_1$, but we have already shown that $p_0D_0 - p_1D_1 = 0.003\ (p_0^{1.94} - p_1^{1.94})$. Substituting the last quantity for pD in formula (12), we have

$$f=\frac{0.062d'^5(p_0^{1.94}-p_1^{1.94})}{w'^2L} \quad . \quad . \quad . \quad . \quad (13)$$

In the Ledoux formula m_0 denotes the weight of steam supplied per second, and in the above formula w' denotes the weight discharged per minute corrected for condensation This formula is essentially of the same character as the Ledoux formula for coefficient of friction (14_L). The formulæ differ only in value of units, coefficients, and correction for condensation, and differ when reduced by an exceedingly small quantity.

By transforming (11) of the paper we obtain

$$w'^2=\frac{20.663pDd'^5}{K\left(1+\frac{3.6}{d'}\right)L}=\frac{20.663pDd'^5}{0.0027\left(1+\frac{3.6}{d'}\right)L}=\frac{7659pDd'^5}{\left(1+\frac{3.6}{d'}\right)L},$$

from which

$$w'=87.45\sqrt{\frac{pDd'^5}{\left(1+\frac{3.6}{d'}\right)L}} \quad . \quad . \quad . \quad . \quad (14)$$

The formula (14) given above resembles very closely the one given by the late George H. Babcock in "Steam," page 89, the difference, in fact, being that Babcock uses a coefficient of 87, whereas the calculation with the value of the coefficient K, as obtained from our experiments, gives a coefficient about one-half per cent. greater. The formula in which this coefficient is used is of a form deduced several years ago by Professor Unwin, and accords very closely with Weisbach's statements of the flow of liquids in pipes. How the coefficient, as given above, was obtained by Mr. Babcock is unknown to the writer; but from its close agreement with the results of our experiments it was probably based on good authority.

The following table calculated by Mr. Babcock is extracted from "Steam":

TABLE OF FLOW OF STEAM THROUGH PIPES.

Initial pressure by gauge. Lbs. per sq. inch.	Diameter of Pipe in Inches. Length of Each = 240 Diameters.													
	¼	1	1¼	2	2½	3	4	5	6	8	10	12	15	18
	Weight of steam per minute in pounds, with one pound loss of pressure.													
1...	1.16	2.07	5.7	10.27	15.45	25.28	46.85	77.3	115.9	211.4	341.1	562.4	894	1,177
10...	1.44	2.57	7.1	12.72	19.15	31.45	58.05	95.8	143.6	262.0	422.7	622.5	986	1.458
20...	1.70	3.02	8.3	14.94	22.49	36.94	68.20	112.6	168.7	307.8	496.5	731.3	1,170	1,713
30...	1.91	3.40	9.4	16.84	25.35	41.65	76.84	126.9	190.1	346.8	559.5	824.1	1,318	1,930
40...	2.10	3.74	10.3	18.51	27.87	45.76	84.45	139.5	209.0	381.3	615.3	906.0	1,450	2,122
50...	2.27	4.04	11.2	20.01	30.13	49.48	91.34	150.8	226.0	412.2	665.0	979.5	1,567	2,294
60...	2.43	4.32	11.9	21.38	32.19	52.87	97.60	161.1	241.5	440.5	710.6	1,046.7	1,675	2,451
70...	2.57	4.58	12 6	22.65	34.10	56.00	103.37	170.7	255.8	466.5	752.7	1,108.5	1.774	2,596
80...	2.71	4.82	13.3	23.82	35.87	58.91	108.74	179.5	269.0	490.7	791.7	1,166.1	1,866	2,731
90...	2.83	5.04	13.9	24.91	37.52	61.62	113.74	187.8	281.4	513.3	828.1	1,219.8	1,951	2,856
100...	2 95	5.25	14.5	25.96	39.07	64.18	118.47	195.6	293.1	534.6	862.6	1,270.1	2,032	2,975
120 ..	3.16	5 63	15.5	27.85	41.93	68.87	127.12	209 9	314.5	573.7	925.6	1,363.3	2.181	3,193
150...	3.45	6.14	17.0	30.37	45.72	75.09	138.61	228.8	343.0	625.5	1,009.2	1,486.5	2 378	3,481

By similar reduction from equation (11) we can obtain the following formula for the diameter of pipe expressed in inches:

$$d' = \sqrt[5]{\frac{w'^2\left(1 + \frac{3.6}{d'}\right)L}{7654pD}} = 0.167\sqrt[5]{\frac{w'^2\left(1 + \frac{3.6}{d'}\right)L}{pD}} \quad (15)$$

For a 6-inch pipe $\left(1 + \frac{3.6}{d}\right) = 1.6$, which, substituted in the foregoing formula and reduced, makes

$$d' = 0.184\sqrt[5]{\frac{w'^2 L}{pD}} \quad . \quad . \quad . \quad . \quad . \quad (16)$$

This formula is very nearly identical with that given by Ledoux when reduced to English units. Thus, for the example considered, and for a flow of 1 kilogram per second, or 132 pounds per minute, the diameter, as given by the above formula, is 4.4 inches, and by the Ledoux formula is 4.39 inches. It would also seem that this formula is accurate enough for practical purposes, since the variation in pipe sizes as computed by this formula, or by the more complicated ones, is slight and not of practical importance. The writer also believes that this formula, which differs from that given by Ledoux principally by the use of the value of density of steam or weight per cubic foot, instead of the difference of exponential function of the pressure, is, on the whole, much more convenient to use than that of Ledoux. To use the above formula the engineer must employ both the table of logarithms and a steam table ; to use the Ledoux formula a table of logarithms alone will suffice, but the computations are more difficult. The table is to be recommended in either case as giving results with sufficient accuracy, and affording great economy of time.

Regarding the question raised by Mr. Kent as to the law of variation of the coefficient of friction, the writer would say that in view of the investigations already quoted in the paper, and especially in view of those made by Professor Unwin in relation to the motion of compressed air, there is little doubt but that the coefficient of friction varies both with change of velocity and with change in diameter of the pipe. The experiments made at Sibley College with small and large pipes clearly indicated a change in coefficient of friction with the diameter, but we were unable to decide in regard to the influence of velocity, as the effect of change of velocity seemed less than the unavoidable errors of observation.

Mr. Kent refers to the variation in results of certain individual experiments from the average. This will also be noted in the experiments made by Ledoux, and serves to indicate the difficulty involved in noting all the conditions which take place at any given time. In some cases it is fair to suppose that mistakes of observation were made, but in other cases it is probable that some important conditions bearing on the subject were unnoted.

The general results of the investigation, and the close agreement of the formula adopted with that given by the late George H. Babcock and by Ledoux, would indicate that the results are

to be relied upon as accurate for practical purposes, although liable to vary to a certain extent with change of conditions, which, no doubt, depend upon the method of erection of the pipe and the amount of moisture entrained in the steam.

Appended to this closure are given abstracts of the data and results taken from the paper by Ledoux.

EXPERIMENTS OF M. LEDOUX.

FLOW OF STEAM IN IRON PIPE 330 METRES (1,082.7 FEET) IN LENGTH; DIAMETER, 75 MILLIMETRES (2.95 INCHES).

No. of Experiment.	Length of Conduit.		Depth Below the Surface.		Initial Press.		Final Press.		Incr. in Press. Due to Weight.	
	Metres.	Feet.	Metres.	Feet.	Atmos.	Lbs. per sq. in.	Atmos.	Lbs. per sq. in.	Atmos.	Lbs. per sq. in.
A	130	426.5	100	328.1			2.62	38.50	0.01	0.15
	230	754.6	200	656.2	2.96	43.50	2.44	35.86	0.03	0.44
	330	1,082.7	239	784.1			2.18	32.04	0.03	0.44
B	130	426.5	100	328.1			3.81	56.00	0.02	0.29
	230	754.6	200	656.2	3.88	57.02	3.66	53.78	0.04	0.59
	330	1,082.7	239	784.1			3.60	52.90	0.05	0.73
C	130	426.5	100	328.1			5.44	79.95	0.03	0.44
	230	754.6	200	656.2	5.89	86.56	4.99	73.34	0.07	1.02
	330	1,082.7	239	784.1			4.60	67.60	0.07	1.03
D	130	426.5	100	328.1			4.52	66.42	0.03	0.44
	230	754.6	200	656.2	5.78	84.94	3.48	51.14	0.05	0.73
	330	1,082.7	239	784.1			2.45	36.00	0.05	0.74
E	130	426.5	100	328.1			4.69	68.82	0.03	0.44
	230	754.6	200	656.2	5.78	84.94	3.90	57.32	0.05	0.73
	330	1,082.7	239	784.1			2.34	49.08	0.05	0.74
F	130	426.5	100	328.1			4.65	68.33	0.03	0.44
	230	754.6	200	656.2	5.76	84.65	3.52	57.73	0.03	0.44
	330	1,082.7	239	784.1			2.29	33.64	0.05	0.73
G	330	1,082.7	239	784.1	5.97	87.74	5.44	79.95	0.07	1.03
H	330	1,082.7	239	784.1	5.97	87.74	5.68	82.75	0.08	1.17
I	330	1,082.7	239	784.1	3.00	44.09	2.69	33.65	0.04	0.59

Area, 7.774 square metres = 828.9 square feet.

FLOW OF STEAM IN IRON PIPE 330 METRES (1,082.7 FEET) IN LENGTH; DIAMETER, 75 MILLIMETRES (2.95 INCHES).

No. of Experiment.	Final Pressure Corrected.		Weight of Steam Supplied.		Wt. of Steam Condensed per Hour.		Values of F.	
	Atmos.	Lbs. per sq. in.	Kilog's per Sec.	Lbs. per Minute.	Kilog's per sq. Metre.	Lbs. per sq. ft.	Observed.	Average.
A ...	2.62	38.50	0.221	27.04	1.05	0.2058	0.00134	0.00119
	2.44	36.06					0.00110	
	2.18	32.03					0.00115	
B ...	3.81	56.09	0.162	21.43	1.19	0.2292	0.00068	0.00100
	3.66	53.78					0.00121	
	3.60	52.90					0.00111	
C....	5.44	79.95	0.362	48.07	1.31	0.2683	0.00125	0.00132
	4.99	73.34					0.00137	
	4.60	67.60					0.00134	
D ...	4.52	66.43	0.538	71.14	1.22	0.2500	0.00142	0.00133
	3.48	51.14					0.00134	
	2.45	35.86					0.00122	
E ...	4.69	68.83	0.545	72.06	1.21	0.2478	0.00123	0.00118
	3.90	58.64					0.00111	
	2.34	34.38					0.00120	
F....	4.65	68.33	0.546	72.18	1.21	0.2478	0.00123	0.00121
	3.52	51.73					0.00127	
	2.29	33.64					0.00120	
G	5.44	79.96	0.254	33.60	1.33	0.2724	0.00123	0.00123
H	5.63	82.74	0.198	26.20	1.34	0.2744	0.00132	0.00132
I.....	2.69	39.53	0.154	20.37	1.09	0.2140	0.00110	0.00110

EXPERIMENTS OF LEDOUX.

Diameter of Pipe, 100 Millimetres (3.94 Inches).

Length, Metres.	Pressure Absolute.		Weight of Steam Supplied per Second, Kilos.	Velocity of Steam.	Coefficient of Friction.	
	Entrance, Kilos.	Discharge, Kilos.			Each Observation.	Mean.
305.8	5.508	3.528			0.00119	
203.3	5.478	4.368	0.908	40.0	0.00114	0.00113
102.3	5.498	4.983			0.00105	
305.8	5.653	4.235			0.00108	
203.3	5.783	4.970	0.842	35.60	0.00099	0.00100
102.3	5.748	5.378			0.00092	
305.8	5.263	2.813			0.00101	
203.3	5.273	3.798	1.036	47.10	0.00102	0.00100
102.3	5.240	4.583			0.00096	
305.8	5.168	3.080			0.00114	
203.3	5.088	3.893	0.911	42.15	0.00106	0.00114
102.3	5.178	4.518			0.00124	
305.8	5.083	4.000			0.00111	
203.3	5.073	4.435	0.698	33.10	0.00102	0.00104
102.3	5.063	4.758			0.00098	
305.8	3.963	2.490			0.00114	
203.3	3.833	2.893	0.690	41.75	0.00111	0.00114
102.3	3.813	3.340			0.00117	
305.8	4.098	2.915			0.00103	
203.3	4.131	3.428	0.673	38.85	0.00097	0.00096
102.3	4.058	3.740			0.00089	
305.8	4.015	2.248			0.00119	
203.3	4.098	3.995	0.723	41.65	0.00125	0.00121
102.3	4.203	3.718			0.00120	
305.8	4.040	1.393			0.00125	
203.3	4.048	2.013	0.692	40.35	0.00127	0.00125
102.3	4.060	2.588			0.00123	
305.8	3.233	1.303			0.00105	
203.3	3.265	2.100	0.690	50.15	0.00121	0.00109
102.3	3.145	2.645			0.00101	
305.8	3.280	1.990			0.00166	
203.3	3.175	2.363	0.486	35.65	0.00163	0.00166
102.3	3.220	2.823			0.00169	
305.8	2.828	1.705			0.00115	
203.3	2.885	2.140	0.508	40.90	0.00123	0.00123
102.3	2.883	2.540			0.00131	

Mean of all results = 0.00115.

EXPERIMENTS OF LEDOUX.

FLOW OF STEAM IN PIPES. DIAMETER OF PIPE, 71 MILLIMETRES (2.79 INCHES).

Length, Metres.	Pressure Absolute.		Weight of Steam Supplied per Second, Kilos.	Velocity of Steam.	Coefficient of Friction.	
	Entrance, Kilos.	Discharge, Kilos.			Each Observation.	Mean.
304.9	5.777	1.449			0.00104	
202.9	5.728	3.500	0.546	45.45	0.00101	0.00097
102.0	5.678	4.818			0.00087	
304.9	5.653	2.578			0.00121	
202.9	5.678	3.953	0.456	38.40	0.00117	0.00110
102.0	5.703	5.075			0.00093	
304.9	5.953	4.060			0.00132	
202.9	5.978	4.875	0.377	30.30	0.00123	0.00116
102.0	5.918	5.513			0.00093	
304.9	4.930	1.475			0.00123	
202.9	4.955	3.068	0.429	41.20	0.00121	0.00115
102.0	4.905	4.200			0.00102	
304.9	5.023	2.32			0.00130	
202.9	4.962	3.44	0.393	37.40	0.00123	0.00120
102.0	4.968	4.360			0.00107	
304.9	5.073	3.268			0.00112	
202.9	4.917	3.858	0.368	35.35	0.00103	0.00100
102.0	4.975	4.555			0.00086	
304.9	4.075	1.360			0.00132	
202.9	4.088	2.568	0.339	38.90	0.00133	0.00126
101.0	4.064	3.478			0.00113	
304.9	3.898	1.923			0.00110	
202.9	3.972	2.763	0.328	38.90	0.00114	0.00103
102.0	3.956	3.54			0.00085	
304.9	4.039	2.640			0.00143	
202.9	4.013	3.130	0.260	31.00	0.00142	0.00134
102.0	4.003	3.648			0.00119	
304.9	3.068	1.508			0.00099	
202.9	3.078	2.090	0.276	41.80	0.00104	0.00100
102.0	3.083	2.658			0.00096	
304.9	3.047	1.808			0.00119	
202.9	3.072	2.315	0.232	35.70	0.00118	0.00112
102.0	3.062	2.755			0.00100	
304.9	3.093	2.153			0.00117	
202.9	3.050	2.473	0.212	32.05	0.00111	0.00105
102.0	3.088	2.870			0.00087	

Mean of all results = 0.00112.

EXPERIMENTS OF LEDOUX.

FLOW OF STEAM IN PIPES. DIAMETER OF PIPE, 47 MILLIMETRES (1.85 INCHES).

Length, Metres.	Pressure Absolute.		Weight of Steam Supplied per Second, Kilos.	Velocity of Steam.	Coefficient of Friction.	
	Entrance, Kilos.	Discharge, Kilos.			Each Observation.	Mean.
303.60	3.325	1.570			0.00100	
202.75	3.211	2.183	0.109	35.75	0.00093	0.00093
101.30	3.208	2.768			0.00085	
303.60	3.168	2.128			0.00071	
202.75	3.093	2.425	0.104	35.10	0.00069	0.00068
101.30	3.063	2.755			0.00064	
303.60	3.122	1.693			0.00080	
202.75	3.117	2.178	0.108	36.35	0.00086	0.00082
101.30	3.117	2.698			0.00081	
303.60	4.125	1.425			0.00086	
202.75	4.105	2.473	0.153	39.86	0.00089	0.00085
101.30	4.083	3.430			0.00079	
303.60	4.068	1.855			0.00084	
202.75	4.028	2.635	0.145	38.45	0.00087	0.00084
101.30	4.048	3.455			0.00081	
303.60	5.048	1.468			0.00099	
202.75	5.071	3.033	0.176	37.90	0.00102	0.00096
101.30	5.060	4.268			0.00089	
303.60	5.058	2.123			0.00088	
202.75	5.078	3.320	0.177	38.15	0.00090	0.00086
101.30	5.083	4.370			0.00080	
303.60	6.045	1.510			0.00104	
202.75	6.028	3.523	0.206	37.50	0.00106	0.00100
101.30	5.963	5.003			0.00091	
303.60	5.926	2.148			0.00101	
202.75	5.928	3.740	0.197	36.35	0.00102	0.00098
101.30	5.931	5.070			0.00091	
303.60	5.953	3.023			0.00097	
202.75	5.935	4.740	0.186	34.50	0.00097	0.00094
101.30	5.940	5.070			0.00086	
202.75	3.117	1.913	0.101	34.05	0.00120	0.00117
101.30	3.129	2.605			0.00114	
202.75	4.078	2.348	0.142	37.25	0.00110	0.00108
101.30	4.093	2.378			0.00105	
202.75	5.063	2.888	0.170	36.50	0.00115	0.00109
101.30	5.053	4.193			0.00103	

Mean of all results = 0.00094.

DCCCIV.*

A PNEUMATIC DISPATCH-TUBE SYSTEM FOR RAPID TRANSPORTATION OF MAILS IN CITIES.

BY B. C. BATCHELLER,† NEW YORK CITY.

THE history of the art of pneumatic transmission practically begins in London in 1853, when the Electric and International Telegraph Company constructed the first tube for dispatching their telegrams. This tube was 1½ inches in diameter and 225 yards in length. It was constructed under the direction of their engineer, Mr. Josiah Latimer Clark.

Five years later a larger tube, 2¼ inches in diameter and 1,340 yards in length, was built by the company. This was the beginning of the pneumatic-tube system in London, and it has grown year by year until a network of tubes radiating from the general postoffice covers the business portion of the city, connecting with all of the sub-postoffices. Most of these tubes are 2¼ inches in diameter, and none is more than 3 inches. Light felt carriers containing telegrams and messages are dispatched through them. Paris was not far behind London in constructing a pneumatic telegraph. They use tubes of about the same diameter, but instead of having them radiate from central stations they are laid in circuits or polygons, the carriers travelling around the circuits in one direction only. Berlin and Vienna, as well as the smaller cities of England, now have systems of small tubes for dispatching telegrams and messages between the main and sub-postoffices. All are familiar with the pneumatic tubes used in our large stores for transmitting cash to and from a

* Presented at the New York meeting (December, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

† This paper was secured by invitation of the committees having charge of the meeting, to explain the operation of the system before the excursion organized to visit the plant in operation at the New York postoffice. It was illustrated by lantern slides and stenographically reported from the verbal delivery. It has been felt that its interest justified its permanent record.

centrally located cashier's desk. The Western Union Telegraph Company has constructed several lines of small tubes between its offices in New York city, it having been found that messages

FIG. 121.

1. Carrier used in the Berlin system. 2. Largest carrier used in the London system. 3. Six-inch carrier used in the first Philadelphia system. 4. Eight-inch carrier used in New York and Boston.

can be dispatched through tubes more quickly than over the wires.

This paper relates to pneumatic tubes of a much larger size than any that are used in Europe—tubes which are not only large enough for the transmission of telegrams, messages, etc., but bulky mail, packages, merchandise, etc. (Fig. 121.)

The Philadelphia Line.

The history of large pneumatic tubes for the transportation of mail began in the city of Philadelphia in the winter of 1892–93.

A company was formed in that city which had for its object the construction and operation of a system of tubes, to be laid under the streets, for the rapid transportation of United States mail, merchandise, messages, packages, etc.

It began by making a contract with the United States Government for the transportation of the mail between the main postoffice and a branch postoffice on Chestnut Street near Third Street, a distance of about 3,000 feet, or little more than half a mile. (Fig. 122.)

The history of this pioneer company is similar to that of many others that launch new enterprises. Unexpected obstacles were encountered which caused annoying delays. It had been supposed that there would be no difficulty in obtaining suitable tubes. An order was placed with a large manufactory of wrought-iron pipe, and it was not until the first delivery was made that the imperfections of this material were known. When the work was well in hand serious defects were discovered in the plans of the terminal apparatus for sending and receiving carriers. I might mention other difficulties that were met by this company in the construction of the first line, but you can imagine what they were, for they are common to all new engineering enterprises.

After being compelled to abandon wrought-iron tubes, it was decided to bore a sufficient quantity of cast-iron water-pipe, and machine the ends so that they would fit evenly. This was considered a makeshift at the time, but no better material than cast iron has since been found for the purpose.

Twelve boring machines were hastily improvised, and in the short space of seven weeks 6,000 feet of 6-inch water-pipe were bored. The finished size of the tubes was $6\frac{1}{8}$ inches inside diameter, in regular 12-foot lengths. The joints were made with lead and oakum calked in the usual manner. The bends in the tube were made of seamless brass tubing filled with rosin and bent to the desired radius, the work of bending being done by a coppersmith. Flanges were screwed to the bends in order to connect them to the iron tubes.

Two tubes were laid for transportation in opposite directions. They were simply buried in the ground, without any special support or covering, one being placed directly above the other.

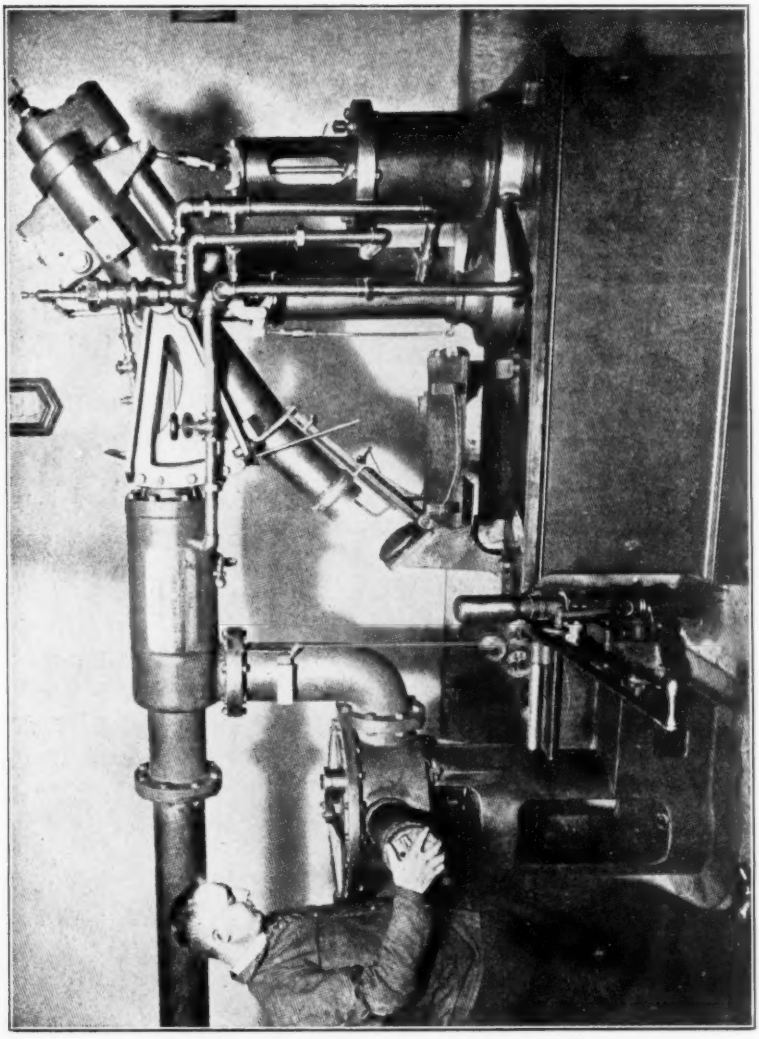


FIG. 122.—SIX-INCH TUBE SENDING AND RECEIVING APPARATUS AT THE BOURSE, PHILADELPHIA.

They were separated by cast-iron brackets. At the main post-office the tubes enter the basement on the Chestnut Street side, extend suspended from the ceiling for 200 feet through a corridor, and terminate on the first floor about the centre of the

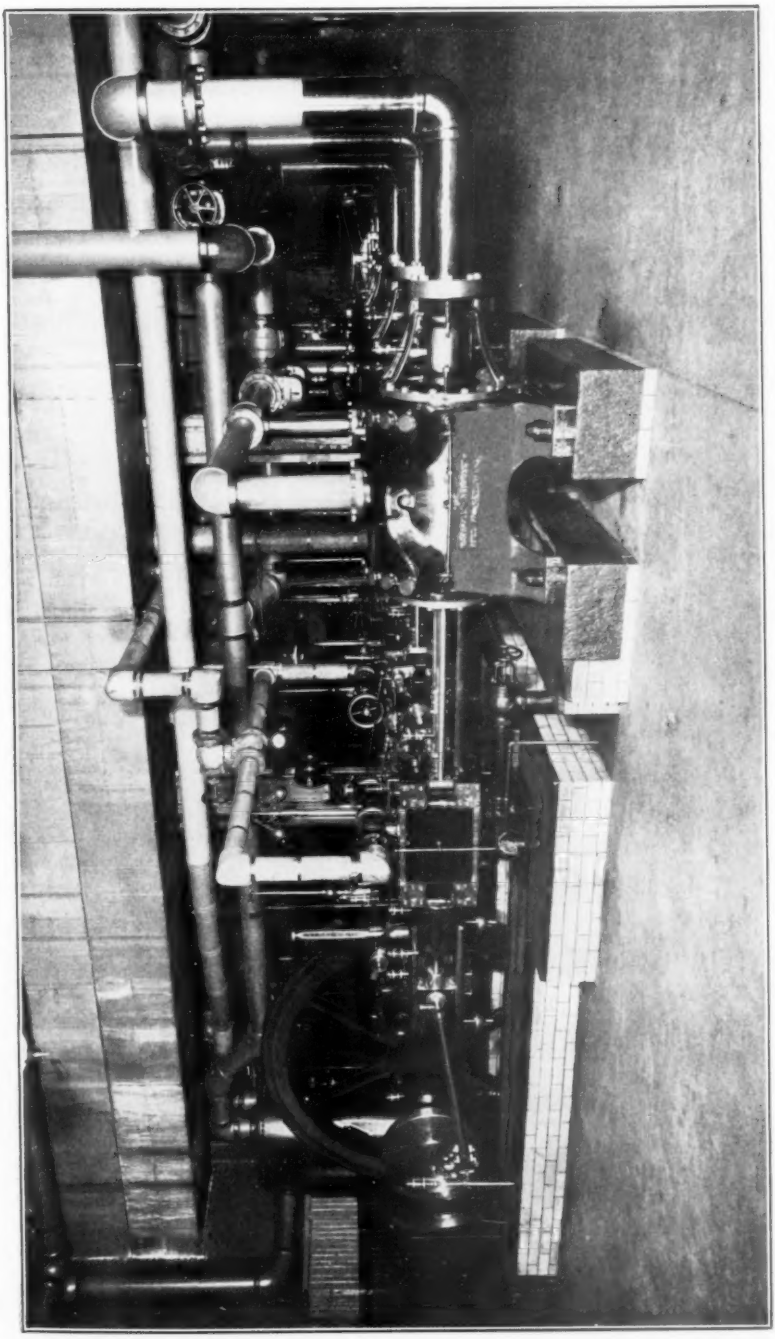


FIG. 123.—AIR COMPRESSOR FOR OPERATING PNEUMATIC TUBES, PHILADELPHIA.

building. At the branch postoffice they also entered the basement and terminated on the first floor, but about a year after the line was opened the branch postoffice was removed from Chestnut Street to the basement of the Bourse Building, where it is now located.

Power for the operation of the system was and is still supplied by an air compressor located in the basement of the main postoffice. The compressor is of the duplex type, having two steam cylinders and two air cylinders arranged with a common crank-shaft and fly-wheel in the centre. The cranks are located between the air and steam cylinders inside a yoke that connects the steam and air piston-rods. The stroke is 24 inches, the diameter of the steam cylinders 10 inches, and the air cylinders 18 inches. The horse-power developed under normal conditions is about 25.

The compressor discharges its air into a tank under a pressure of 6 pounds per square inch. The tank is 4 feet in diameter by 8 feet high, and serves the double purpose of a separator to collect oil and dirt coming from the compressor and a cushion to reduce the pulsations of the compressor.

The air flows from this tank to the transmitter located on the mail floor of the postoffice, thence through one tube to the Bourse, and returns through the other tube to the main postoffice, where it passes through the receiving apparatus, and is then discharged into a second tank, from which the air compressor draws its supply. This tank that receives the air upon its return is open to the atmosphere, in order that any air that has escaped from the tube may be replaced by fresh air drawn into the tank. The tank also serves as a separator to collect any dirt or moisture that may come from the tube.

This arrangement, adopted in Philadelphia five years ago, has been maintained in all systems since constructed, with the exception of the first tank, which has been abandoned because it was found to be unnecessary. The air compressors now pump the air directly into the tubes.

The transmitter or sending apparatus used in Philadelphia is simply a large valve turned by hand. In one position the valve is brought opposite an opening in its casing through which a carrier can be inserted, then the valve is turned until it is in line with the tube, when the air shoots the carrier along in the tube. In the latter position of the valve the opening through

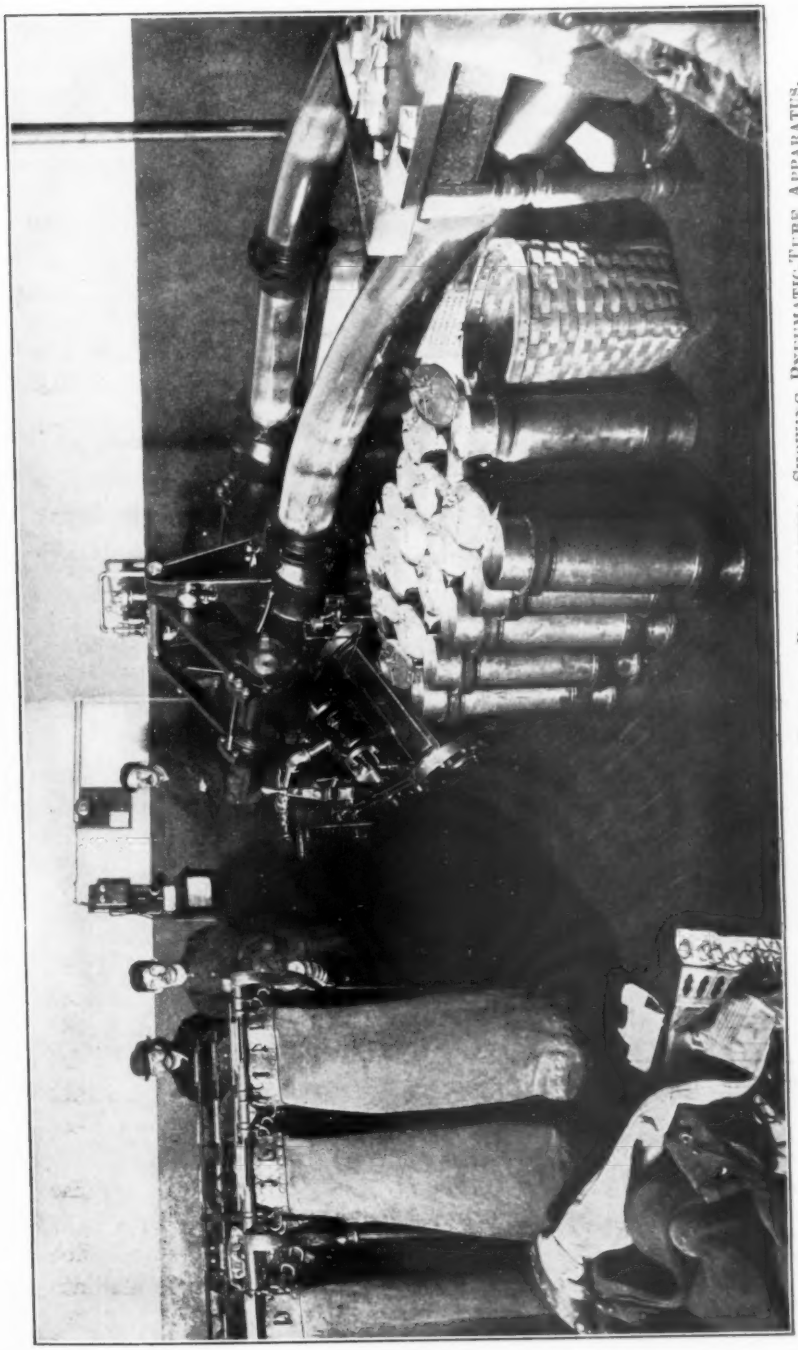


FIG. 124.—MAIL ROOM, PENNSYLVANIA RAILROAD STATION, PHILADELPHIA, SHOWING PNEUMATIC-TUBE APPARATUS.

which the carrier is inserted is closed by a circular plate. When the valve is turned to receive a carrier the air flows around the valve inside its casing, so the continuity of the current is not broken. The receiving apparatus at the main postoffice and at the Bourse is essentially the same as that used in the systems constructed later, which will be described.

The carriers used in this first Philadelphia line are similar but smaller than those used in New York and Boston. They are about 18 inches long by 6⅞ inches inside diameter, and will contain 300 ordinary letters. (Fig. 121.)

Notwithstanding the doubts of some postoffice officials, this tube proved to have ample capacity for transporting all the mail that had to be sent between the two offices, except occasionally some package that was too large to be put into a carrier.

The time of transit of the carriers between the stations is about one minute, which is equivalent to an average speed of about 50 feet per second, or 35 miles per hour, much higher, you will notice, than is used in London, Paris, or Berlin.

Government Contracts.

After the completion of the Philadelphia line nothing was done to extend the pneumatic postal service in this country for a period of nearly four years. This may perhaps be explained by the financial depression that prevailed during this time Gradually, however, the officials of the Postoffice Department came to appreciate the great advantages of such a system, and last year Congress appropriated $150,000 for extensions in other cities. Of this amount New York came in for the largest share. Contracts were made for five separate lines of 8-inch tubes (you will note that the diameter of the tubes was increased two inches over that of the Philadelphia line): Two to be built in New York —one between the main postoffice and the branch postoffice in the Produce Exchange, the other between the main postoffice and branch postoffice "H," near the Grand Central Depot, with three intermediate stations on the line at branch postoffices "D," "F," and "Madison Square."

Another line between the main postoffice, New York, and the main postoffice, Brooklyn, by way of the Brooklyn Bridge.

The fourth line in Philadelphia, between the main postoffice and the Pennsylvania Railroad station, with an intermediate station at the Reading Railway terminal.

And the fifth line in the city of Boston, connecting the main postoffice with the Union Railway station.

The United States may have been a little slow in adopting a pneumatic postal service, but when she did it was done on a scale that is a lesson to her European neighbors. European cities have pneumatic message systems, but the United States is the only country that has a pneumatic postal system. It is the only country that transports its mails by pneumatic tubes. England, France, and Germany are satisfied with 2 and 3-inch tubes, but the United States builds 8-inch.

At the present time all of the above-mentioned lines in New York, Philadelphia, and Boston are in successful operation. The Produce Exchange line in New York was formally opened October 7, 1897; the Boston line, December 17, 1897; the Grand Central line, February 11, 1898; the line to the Pennsylvania station, Philadelphia, April 7, 1898; and the Brooklyn Bridge line, August 1, 1898. The aggregate length of all these double lines is nearly 7½ miles, or 15 miles of tube. To operate them 12 duplex air compressors are required, besides a large amount of terminal apparatus for dispatching and receiving the carriers. To manufacture all of this material, except the air compressors, a factory was constructed at Philadelphia fitted with special machinery.

Tubes.

The tubes are of cast iron, made in 12-foot lengths, with bells cast upon one end. They resemble ordinary water-pipe, but are somewhat thicker and made of a better quality of iron. They weigh before being bored about 720 pounds, or 60 pounds per foot, and are $\frac{9}{16}$ of an inch thick after being bored. They are cast about 8 inches inside diameter, and are bored to a finished diameter of $8\frac{1}{8}$ inches. After boring, a counter-bore is turned at the bottom of the bell, and the male end of the tube is turned to fit the counter-bore of the next length.

After each tube is finished it is gauged by passing a hardened steel gauge mandrel through it. The maximum variation in diameter of bore is about 0.01 inch, and this is principally due to the variations of temperature of the tube during the process of boring. If there is a hard spot in the iron, that part of the tube will become heated and expand, to contract again after the bore is finished.

It has been found necessary to make the cutters slightly larger than the finished diameter of the tubes.

Brass Bends.

Short bends in the tubes are made of brass from seamless tubing bent to a uniform radius of twelve times the diameter of the tube, or eight feet for an 8-inch tube. In order to bend the brass tubing and maintain a uniform cross-section, it is first filled with rosin which is poured in hot and allowed to harden, then the bending is done between a series of rolls.

After the tubes are bent the rosin is melted out; they are cleaned, and brass flanges attached to the ends. This sounds like a very simple process, but a large amount of experience was necessary to produce satisfactory bent tubes.

The brass bends are slightly larger in diameter than the straight cast-iron tube, to allow the carriers to pass freely through them. The slight leakage of air past the carrier while it is passing through the bends is not sufficient to interfere with the proper operation of the system. Since the brass tube is larger in diameter than the iron, tapering connecting pieces must be used to connect the two together in order to provide a smooth, unobstructed surface for the carriers to slide over. For reasons of economy the brass tubes are made as short as the angle will allow, but we are compelled to have a short portion of a straight tube on each end of a bend, it being impossible to bend a tube all the way to its end, for the "bending moment" would be infinite, and, of course, the material will not stand an approach anywhere near this limit. When the tangent point must be at the end of the tube we cut off the unbent portion.

Terminal Apparatus.

At each station, which may be the end or an intermediate point on the line, apparatus is required for sending and receiving carriers.

The sending apparatus, or transmitter as we sometimes term it, consists of two sections of the tube supported in a swinging frame (Fig. 125), so arranged that either section can be brought into line with the main tube, in which a current of air is constantly flowing. One of these tube sections maintains the continuity of the main tube, while the other is swung to one side to receive a

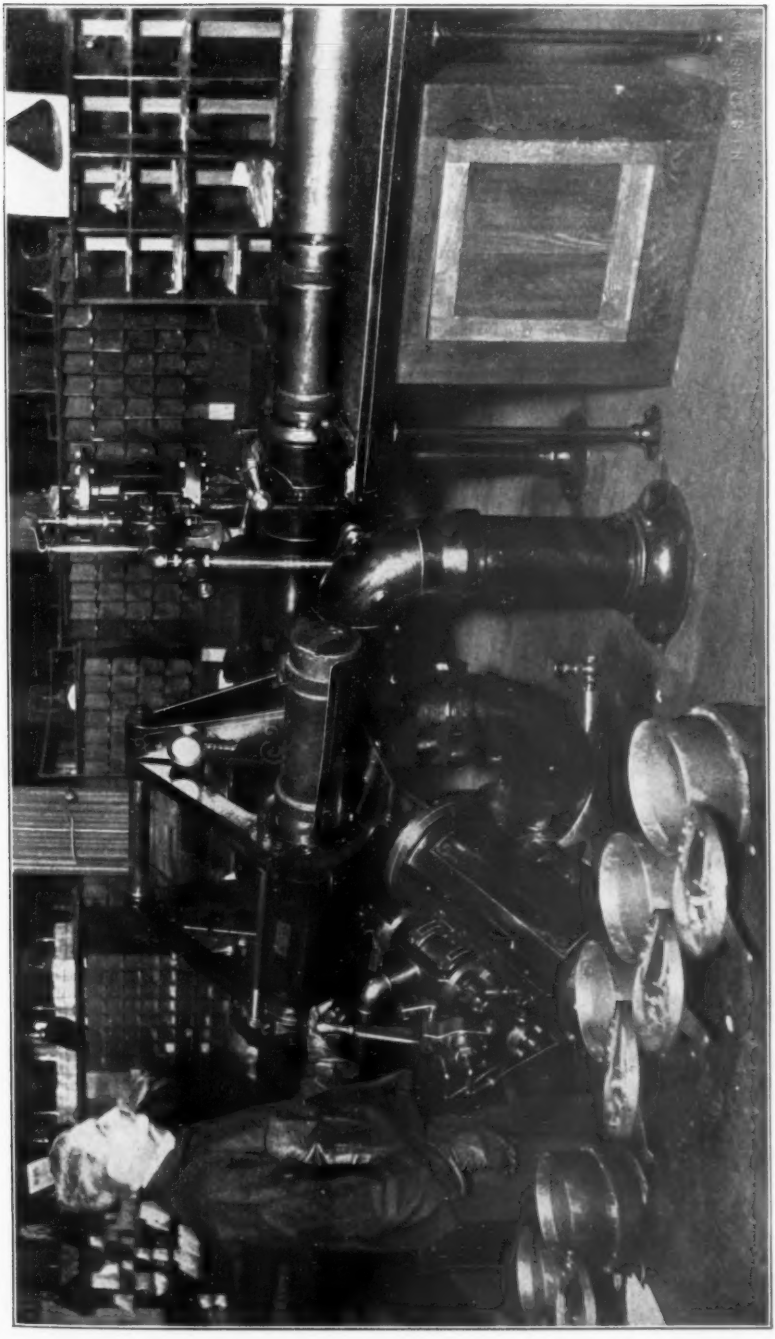


FIG. 125.—PNEUMATIC-TUBE SENDING AND RECEIVING APPARATUS, GENERAL POSTOFFICE, BOSTON, MASS.

carrier. In dispatching, a carrier is placed in an iron trough and then pushed into the open-tube section. The frame carrying the two-tube sections swings until the section containing the carrier is brought in line with the main tube, when the carrier is swept along with the current of air. While the frame is swinging from one position to the other, the air is prevented from escaping by plates that cover the ends of the tube, and a by-pass is provided so that the current is not interrupted. An air motor, consisting of a cylinder and piston, furnishes the power to swing the frame, the operator having simply to move a valve by pulling a lever. When the controlling lever is pulled and latched the frame swings, and as the carrier passes out of the apparatus it trips the lever and the frame swings back automatically into position to receive another carrier.

A certain headway must be given to each carrier in order to prevent them from overtaking each other in the tube, and to give the receiving apparatus at the other end of the line time to act. To prevent carriers being dispatched too frequently, a time-lock has been attached to the sending apparatus, that locks the controlling valve when a carrier is dispatched and keeps it locked for a given period of time, varying from five to fifteen or twenty seconds, according to the adjustment of the lock. The time-locks consist of a vertical cylinder and piston with a spring that constantly tends to force the piston to the bottom of the cylinder. The cylinder is filled with oil and closed at the top, forming what is commonly known as a dash-pot. When the piston is pulled upward the oil flows through holes in the piston that are closed by a valve when it moves downward. A by-pass from one end of the cylinder to the other allows the oil to flow around the piston during its downward motion. This by-pass can be throttled more or less by a small regulating valve, thus controlling the speed and time of descent of the piston. When a carrier is dispatched the piston is pulled up and allowed to descend under the force of the spring. When the piston is pulled up a spring bolt locks the controlling valve, and it remains locked until the piston returns to the bottom of the cylinder and withdraws the bolt. The period of time that the apparatus is locked depends upon the time of descent of the piston in the cylinder of oil, and this can be regulated by opening or closing the by-pass.

Should the attendant attempt to dispatch a carrier before the

locking period has expired the controlling lever will move as if the valve were not locked, but the valve itself will not move, the two being connected by a spring. When the locking period has expired this spring that connects the lever with the valve will move the valve, even though the attendant may be elsewhere attending to other duties. By this automatic arrangement the attendant is not kept waiting for the transmitter to be unlocked.

Open Receiver.

When a carrier arrives at a station its velocity must be checked, and then it must be discharged from the tube. This is the function of the receiving apparatus, which operates automatically. Three forms of receivers are used, and they are known as the open receiver, the closed receiver, and the intermediate station receiver.

The open receiver (Fig. 126) is used where the pressure in the tube is only an ounce or two above the atmosphere, which occurs at the end of the tube farthest removed from the air compressor, and that may be in the same building with the compressor when the outgoing and incoming lines are both operated from one station, as is done with the line in Boston.

The open receiver consists of an air-cushion formed by closing the end of the tube with a sluice-gate and allowing the air to flow out into a branch pipe through slots in the tube located about four feet in the rear of the sluice-gate. When a carrier arrives it passes over the slots, enters the air-cushion, and is brought quickly to rest without injury or shock. As it enters the air-cushion its momentum is destroyed by compressing the air, and this increase of pressure is utilized to move a small valve that admits air to a cylinder located just above the sluice-gate. The piston in this cylinder is attached to the sluice-gate, so that as soon as the carrier comes to rest the gate is raised and there is just pressure enough in the tube to push the carrier out onto a table. When the carrier passes out it raises a finger that in turn moves the valve, and the air pressure acting on the piston in the cylinder closes the sluice-gate. It takes some time to describe this, but the operation occurs very quickly—in less than one second.

Closed Receiver.

With an air pressure in the tube of several pounds above the atmosphere we cannot use the open receiver, because when the

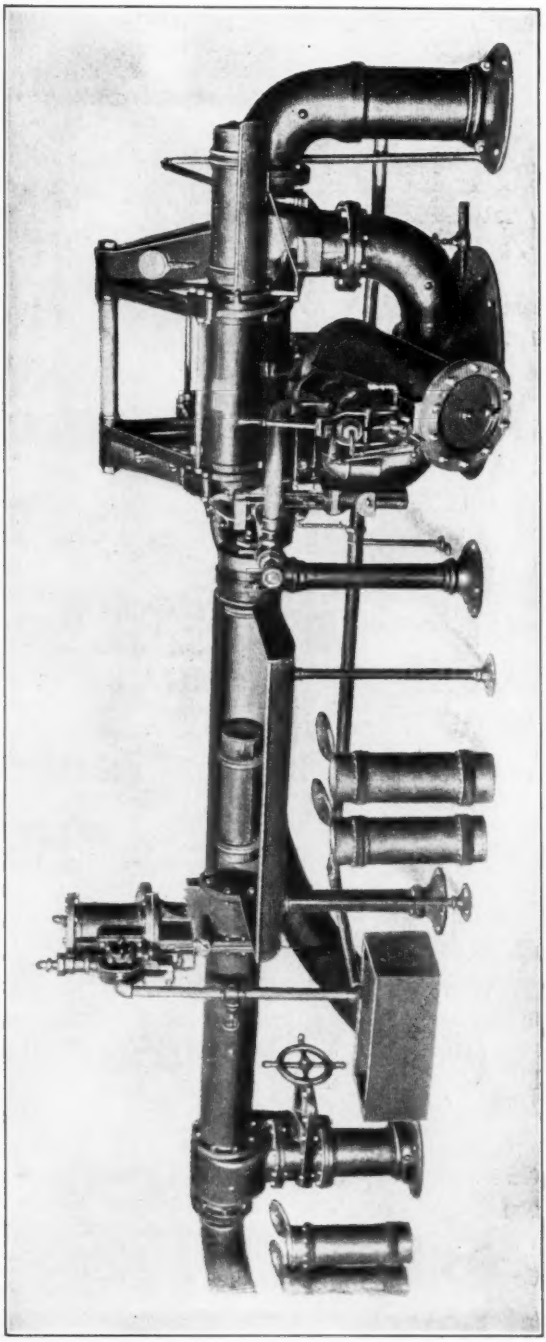


FIG. 136.—SENDER AND OPEN RECEIVER, PRODUCE EXCHANGE LINE, GENERAL POSTOFFICE, NEW YORK CITY.

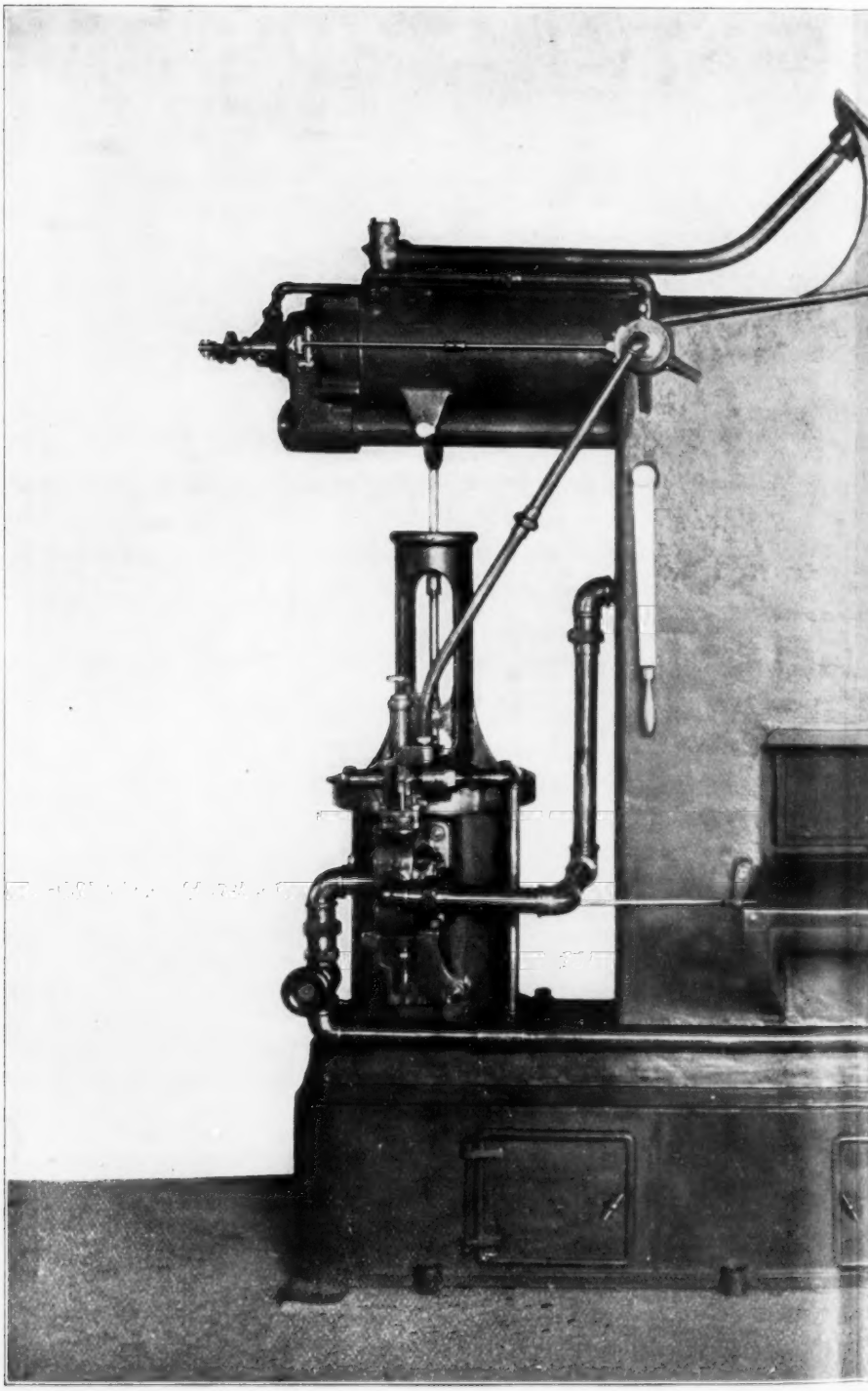
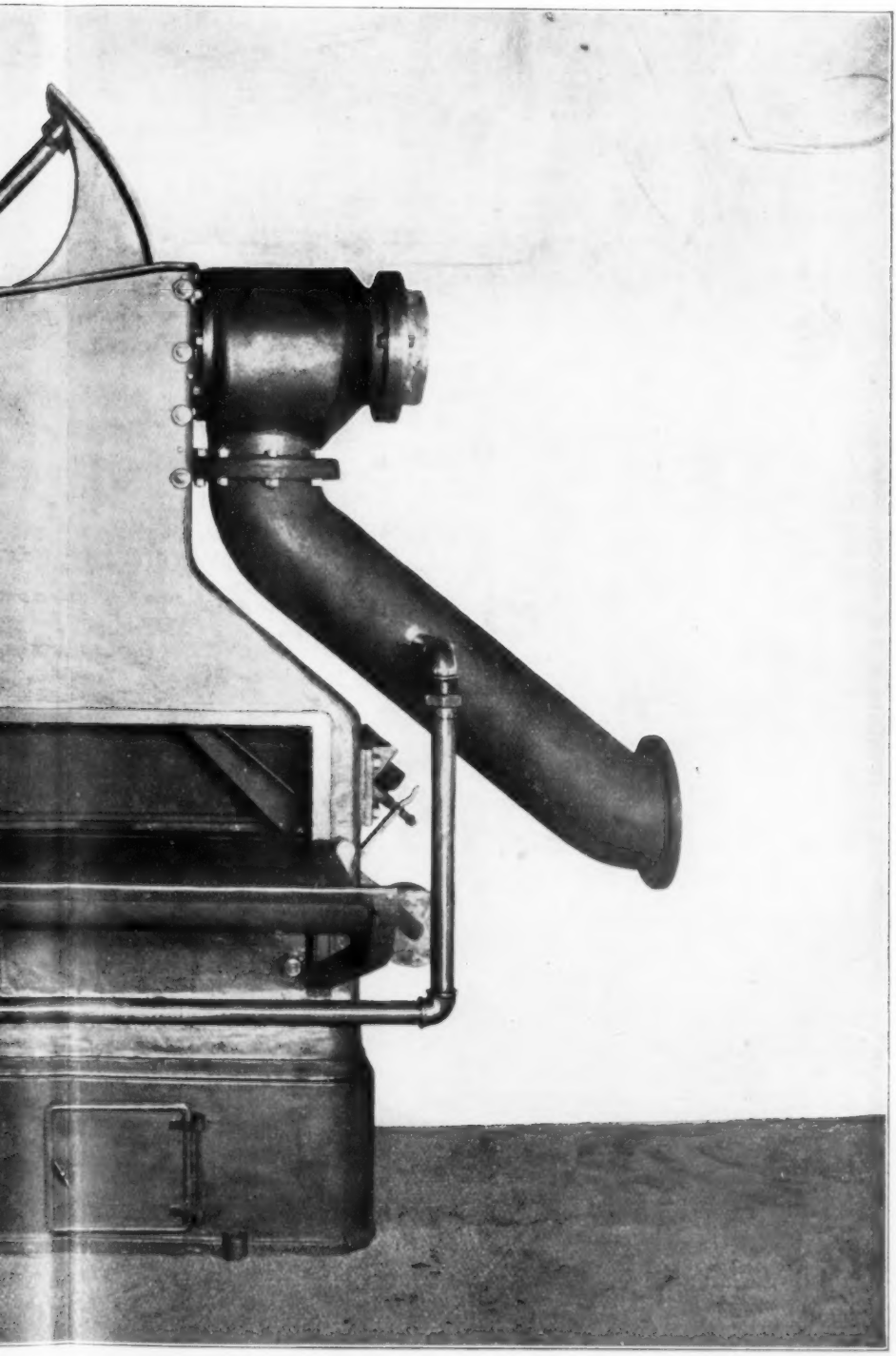


FIG. 127.—CLOSE



V.—Closed Receiver.

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sluice-gate is raised the air would come out with a terrific blast and a roar that could not be permitted; besides, the pressure would shoot the carrier out with great force. For terminal stations where the air current arrives in one tube and returns by the other tube, such as the Union Railway station in Boston,

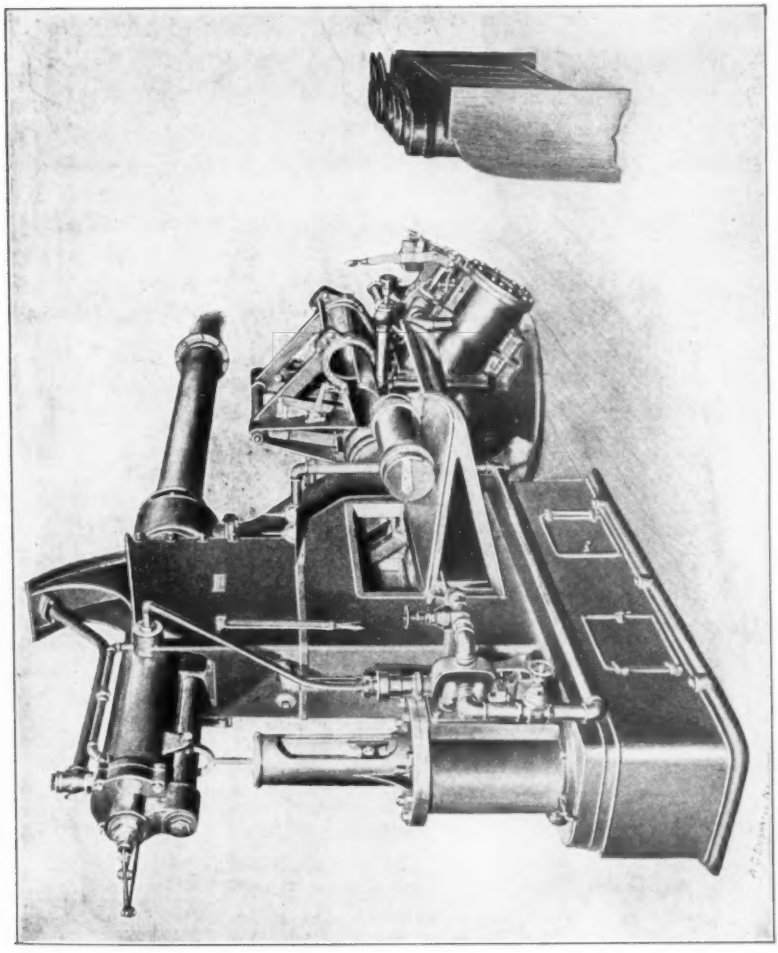


FIG. 128.—SENDER AND CLOSED RECEIVER, PRODUCE EXCHANGE LINE, POSTAL STATION "P," NEW YORK CITY.

we have what we term a closed receiver (Figs. 127 and 128). It consists of a section of tube closed at one end, forming an air-cushion, which we will call the receiving chamber. This receiving chamber is a prolongation of the tube. As carriers arrive they run directly into it and come to rest without shock by com-

pressing the air. The air current flows out of the tube through a branch located close to the receiving chamber that leads to the transmitter of the return line. The receiving chamber is mounted upon trunnions and has one end connected to a piston in a cylinder so arranged that the movement of the piston tilts the receiving chamber up at an angle of 40 degrees to discharge the carrier and then returns it to a horizontal position. While it is tilted the end of the tube is covered by a circular plate attached to the receiving chamber. The air pressure is used to move the piston and receiving chamber, the movement being controlled by a small piston slide-valve, as in the open receiver.

The slide-valve is moved by the increase of air pressure in the receiving chamber when a carrier arrives. It is returned to its normal position by the weight of the carrier, when the carrier slides out of the receiving chamber onto a pivoted cradle. The cradle is connected mechanically to the valve stem and swings down into a horizontal position under the weight of the carrier, thereby moving the valve.

Intermediate-Station Receiver.

I have already stated that the line between the General Postoffice, New York, and the Grand Central Railroad depot includes three intermediate stations (Fig. 129). These stations are not located on branches off the line, but the main line is carried in and out of each station, so that the intermediate stations are located directly on the main line. Automatic receiving apparatus is used in these stations, whereby a carrier destined for some particular station on the line can be dispatched from the General Postoffice, and is so arranged that such a carrier will enter each intermediate station and pass out again until it arrives at the station for which it was destined, where it will be automatically discharged from the tube. The station at which the carrier will be discharged is determined by the diameter of a flat circular metal disc placed on the front of the carrier. These discs are made in graduated sizes, corresponding with the apparatus in the various stations on the line. The disc serves the purpose of closing an electric circuit, which it cannot do until it arrives at a station where the size of the disc corresponds with the distance between two electric contact points.

The intermediate-station receiver is of the "closed receiver"

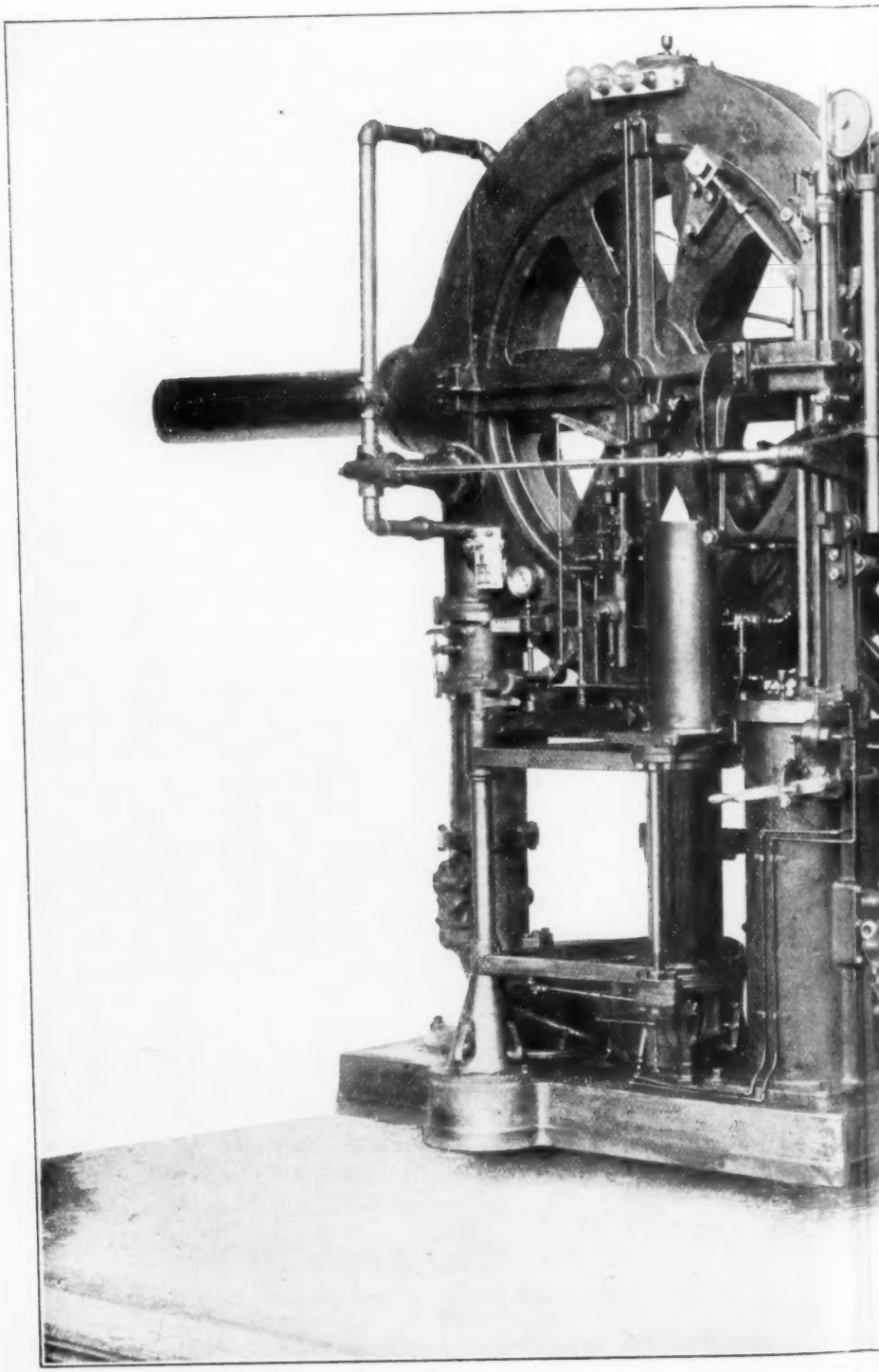
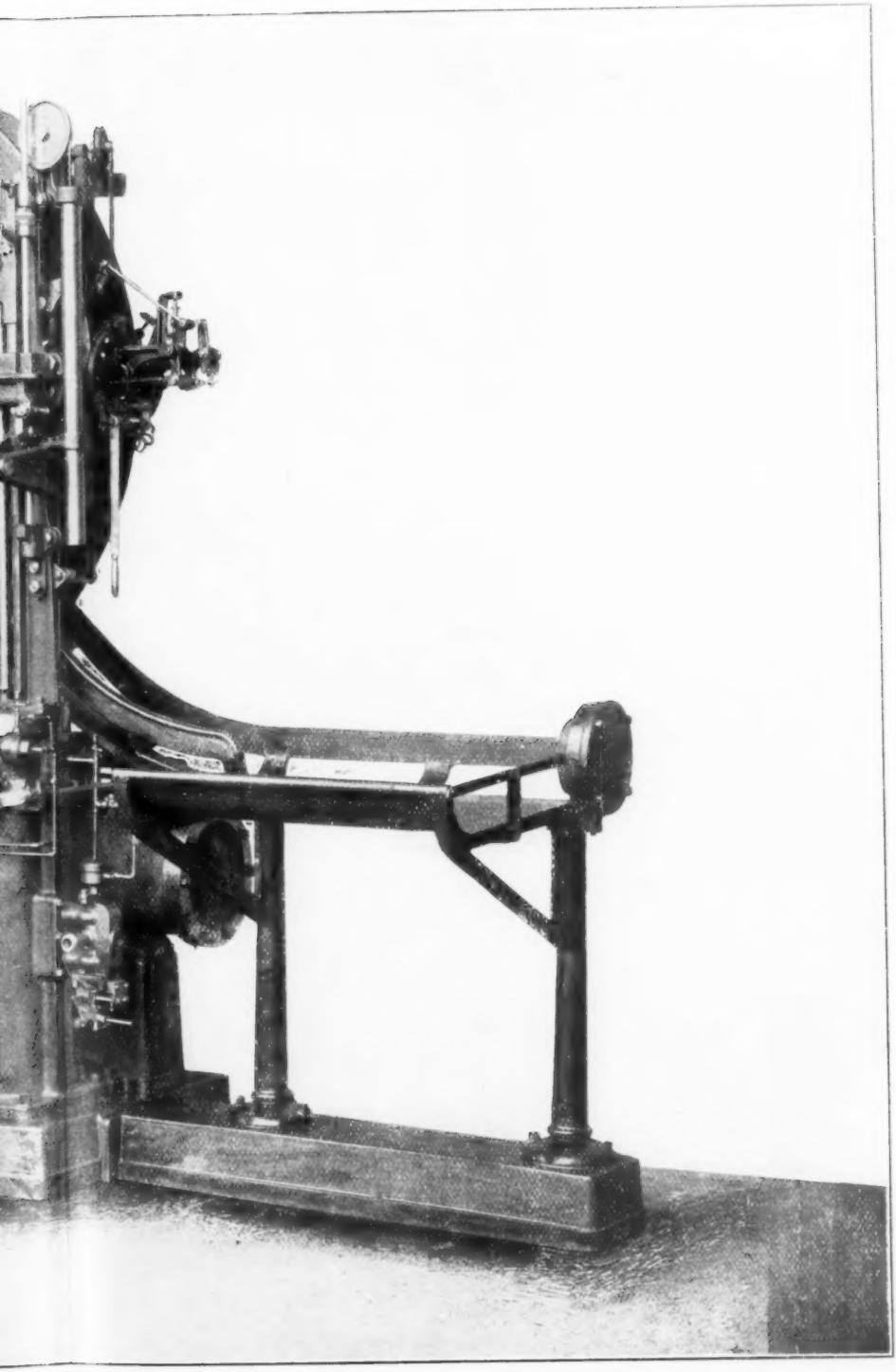


FIG. 129.—INTERMEDIATE-STAT



ATE-STATION SENDER AND RECEIVER.

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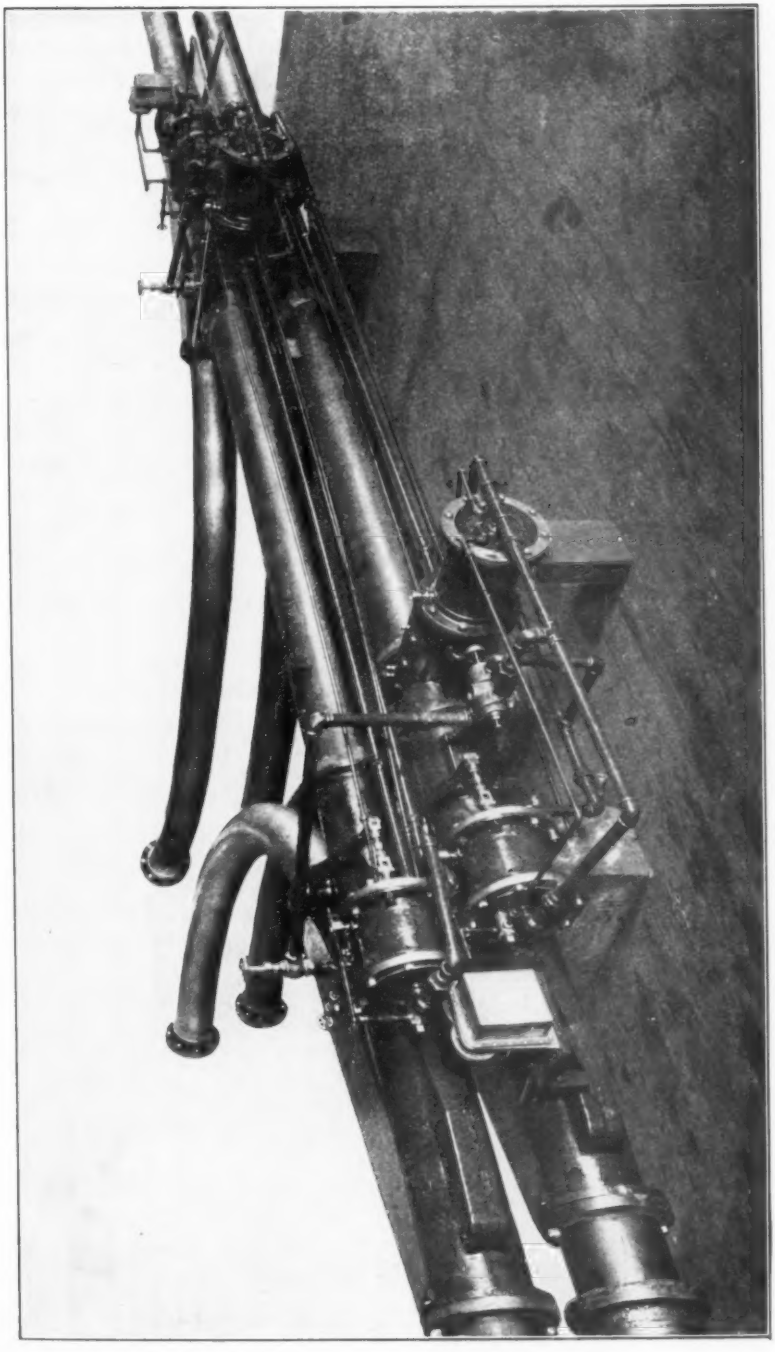

FIG. 130.—A SET OF CUT-OUT SWITCHES.

type, consisting of a chamber that forms a prolongation of the tube and an air-cushion to stop the carriers without shock. This cylindrical chamber is located in the horizontal diameter of a wheel that can be rotated through an angle of 45 degrees to discharge a carrier onto a table, or through an angle of 90 degrees to redispatch a carrier into the tube and on to the next station. When the wheel rotates its broad flat rim covers the end of the tube, thereby preventing the escape of air. The wheel is rotated by a vertical cylinder and piston, using the air pressure in the tube as a motive power. When the carrier arrives at the head of the receiving chamber it touches two electric contact points. These contact points are the ends of two insulated needles that project into the chamber, and are held there by springs which allow the needles to retract when the end of the carrier strikes them. The two needles are set at graduated distances apart at the several stations on a line corresponding with the graduated diameter of the discs carried on the front end of the carriers. When a carrier enters the receiver and strikes against the points of the needles, if the disc of the carrier is large enough to span the distance between the two needles, then an electric circuit is closed through the needles and disc, exciting an electro-magnet on the machine that causes a stop to be thrown into the path of the wheel, stopping it at the 45-degree angle, at the same time drawing a slide and opening the receiving chamber so that the carrier can slide out on the table. Had there been no disc on the carrier, or had the disc been too small in diameter to span the space between the needles, then the circuit would not have been closed and the wheel would have rotated through an angle of 90 degrees until the receiving chamber stood in a vertical position in line with a continuation of the tube. The carrier would then drop into the tube and go on to the next station. The air current flows through a by-pass around the wheel. At the first intermediate station on a line the needles are placed farthest apart, requiring the largest size disc to throw the carrier out at that station. At the second station the needles are set nearer together, and at the third station still nearer, and so on, the last intermediate station requiring the smallest disc.

Directly under the receiver and supported upon the same base is a sender for dispatching carriers from the intermediate station. It is similar in all respects to the sender already

described and used in other stations, except that the swinging section of tube swings about a vertical instead of a horizontal axis.

Each intermediate station is arranged to be cut out whenever it is deemed desirable. In order to accomplish this, a vault is built in the street near the station, in which is located a set of cut-out switches or by-pass. (Fig. 130.) These switches are

FIG. 131.—8-INCH CARRIER.

operated electrically from the station near by. The electric current is used to open a small valve. The air pressure furnishes the power to move the switches.

Carriers.

I have already spoken of the carriers which contain the mail and packages during transit through the tube. They are thin steel cylinders, closed at the front end by a convex disc of the same material, carrying a buffer of felt and leather. The rear

end is closed by a hinged lid, secured by a lock. The lock on the lid consists of three radial bolts that pass through the shell of the carrier and the flange of the lid; they are moved by three eccentrics mounted on a common shaft that is turned by a lever on the outside of the lid. (Fig. 131.) The eccentric shaft is located eccentrically on the lid, so that when the handle is turned in the unlocked position it projects beyond the periphery of the carrier. In the locked position it lies within the circumference of the lid. When the carrier is in the tube the lid cannot become unlocked, for the handle cannot turn without striking the walls of the tube.

The shell of the carrier is 24 inches long and 7 inches in diameter. It is surrounded by two bearing rings of woven cotton fabric, specially prepared, and clamped between metal rings. These bearing rings form the wearing surface as the carrier slides through the tube. When they are new they fit the tube closely, but they become worn until they are a quarter of an inch smaller in diameter than the tube. A carrier will run from four to five thousand miles before these rings have to be renewed. Considerable air escapes past the carriers when the rings are worn small, but it affects their velocity very little and does not interfere with the proper working of the system.

An empty carrier weighs about 13 pounds, and when filled with mail 25 or 30 pounds, sometimes more.

The weight of our carriers is a very important factor in the operation of the system, for the life of the bearing rings is inversely proportional to it, and the energy stored in the carriers by virtue of their velocity is directly proportional to it, which means that the lighter the carrier the less is the wear and tear on the whole system.

The energy of a carrier weighing 30 pounds and moving with a velocity of 50 feet per second is 1,165 "foot-pounds," sufficient to do considerable damage if one should strike a solid object. If the end of the tube were turned upward and the carrier allowed to come out freely it would rise nearly forty feet into the air.

Laying Tubes.

Having thus briefly described the special apparatus used, let us turn our attention for a few moments to the work of construction in the streets.

Of course, it is desirable that the tubes should be laid in as nearly a straight line as possible, supported upon a firm foundation, and protected from injury by displacement or frost. To lay them in a straight line for any considerable distance in the streets of our cities is an impossibility. The streets themselves are frequently not straight, the grade of the streets is constantly changing, and our tubes must occupy space not already appropriated for other purposes, if such space can be found. Most of you are more or less familiar with the underground constructions in the streets of our large cities. Almost every foot of space is occupied by sewers, water pipes, steam pipes, telegraph, telephone, and electric-lighting conduits, police- and fire-alarm conduits, trolley-railway conduits, house connections, and last, but not least, subways. Then bear in mind that all these pipes and conduits must be made accessible by frequent manholes that occupy a greater space in the cross-section of a street than the pipe or conduit itself. Now we must find space for our pneumatic tubes, approaching as near a straight line as possible. We are frequently compelled to go deep to avoid other constructions; for example, the tubes recently laid in this city are from three to ten feet below the surface of the street. The depth and direction of the line can be varied by making a slight angle at each joint of not more than 48 minutes, or a deflection of 2 inches in 12 feet, which is usually sufficient to avoid obstructions when the trench is opened half a block in advance. A cast-iron mandrel is used in laying the tubes to insure smooth joints, and the ability to pass this mandrel through the joints limits the amount of deflection that can be made. The mandrel is 18 inches long, one sixty-fourth inch smaller than the bore of the tubes at the centre, and one-eighth inch smaller at each end. It is attached to the end of a long rod and passed through each joint when the tube is set in position, and again after the joint is cast and calked.

The joints are made with yarn and lead, the same as in water and gas pipes. Special care must be taken to insure the entrance of the male end of the tubes into the counter-bores. Each length of tube must be thoroughly cleaned on the inside before it is put into the trench. The tubes are supported upon blocking in the trench and the earth is firmly rammed about them. No other support is used unless the earth is loose, in which case brick piers are built up from solid ground. The brass

bends are surrounded with eight inches of concrete or brick grouted with cement to protect them from injury. They are only one-eighth of an inch thick and easily injured by a pick or shovel if left exposed. The iron tubes are coated on the exterior with tar to protect them from oxidation.

When the tubes are laid in the ground no provision is made for expansion and contraction other than the yielding of the lead joints, but in the lines that have been laid across the Brooklyn Bridge special telescopic joints are provided, for the motion there is about fourteen inches. We have also to provide ball-and-socket joints to allow a slight lateral motion where the ends of the bridge join in the centre.

It is customary to place drip-pots at all low points in the lines to collect any water that may accumulate. These drip-pots have a blow-off pipe leading up to a surface box.

In order to locate leaks and obstructions in the tube a hole is tapped every three hundred feet, and a small pipe is screwed in and led up to the surface of the street, terminating in a surface box. Thus provision is made for attaching a pressure-gauge at any time without digging up the street. The pressure, if noted simultaneously at several points along the tube, is a guide in locating faults.

After the construction of a line is completed the time arrives for sending the first carrier through a tube, and this is an event of great concern for all who have taken any part in the work, especially for the engineers who have been in charge of laying the tubes and for the contractors who have done the work. Even the capitalists who have invested their money in the pipe so securely buried under ground do not escape a feeling of unrest until the first carrier has made its appearance at the far end of the tube. Then, to make doubly sure that there are no obstructions, they usually ask to have the second dispatched around the circuit.

It would be a wonder if during the weeks that the work of laying the tubes is in progress some careless workman did not leave a piece of wood or yarn in the tube to be brought out by the first carrier. When the first line was opened in New York a piece of wood about 3 feet long came out in front of the carrier, but, fortunately, without serious injury to either the carrier or the apparatus. At the opening of the Boston circuit a bundle of

yarn, such as is used in making the joints of the tube, was the only obstacle in the path of the carrier.

Size of Tubes.

The question is frequently asked, "How large a tube can be constructed?" The reply is that it depends entirely upon circumstances. A 4-foot tube or tunnel railway was built in London, but it proved a failure. For postal service I think an 8-inch tube large enough, and it gives a more rapid service than a larger tube, besides being less expensive to build and operate. If the carriers were much larger they could not be readily handled, they could not be dispatched so frequently, the bearing rings would wear out more quickly, it would not be practicable to dispatch carriers with single letters, the mails would have to stand and accumulate, the difficulty of laying the tubes and of getting into buildings would be greatly increased, etc.

There may be localities where a 10-inch or even 12-inch tube will be demanded. There will be no difficulty in making it operate. It is purely a question of expediency.

For dispatching messages a 2 or 3-inch tube is large enough, as has been demonstrated abroad.

Another question frequently asked, "What is the limit in length that a line can be operated?" For an 8-inch tube I should not advise building lines more than four or five miles in length without a relay station. The distance might be greater with larger tubes, but the expense of operating would be large. I do not think it wise to use pressures above 25 pounds per square inch, and with this pressure the speed will decrease as the length of line increases. When the speed becomes low we have lost the great advantage of pneumatic tubes. The longest line that we have built thus far is about three and one-half miles, between the General Postoffice and Forty-fourth Street, New York.

Lines in Operation.

In Philadelphia there are two tube circuits now in operation. The first is the 6-inch line constructed in 1892, which has already been described at the beginning of this paper.

The second circuit connects the General Postoffice with the Pennsylvania Railway station at Broad and Fifteenth Streets, with an intermediate station at the Reading Railway depot

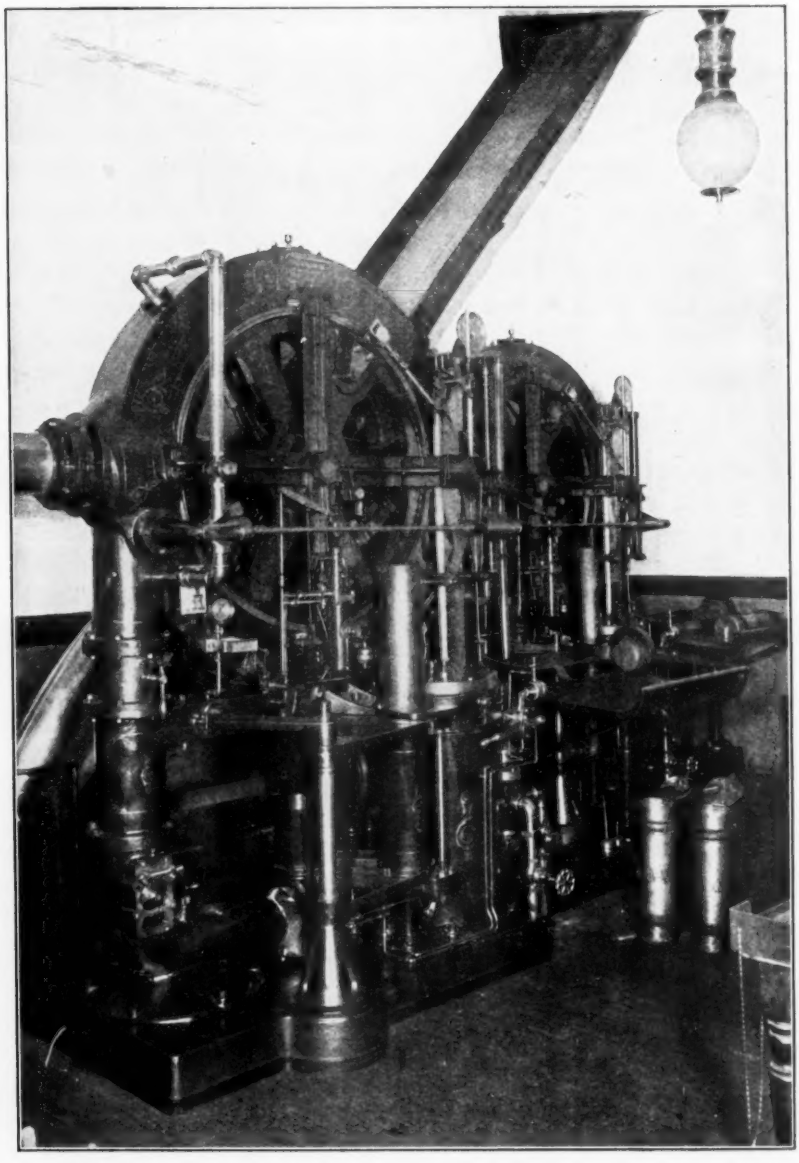

FIG. 132.—MAIL ROOM, READING RAILWAY STATION, PHILADELPHIA, SHOWING PNEUMATIC-TUBE APPARATUS.

known as the Reading Terminal. This circuit consists of a double line of 8-inch tubes laid from the General Postoffice along Tenth Street to Filbert Street, and out Filbert Street to the

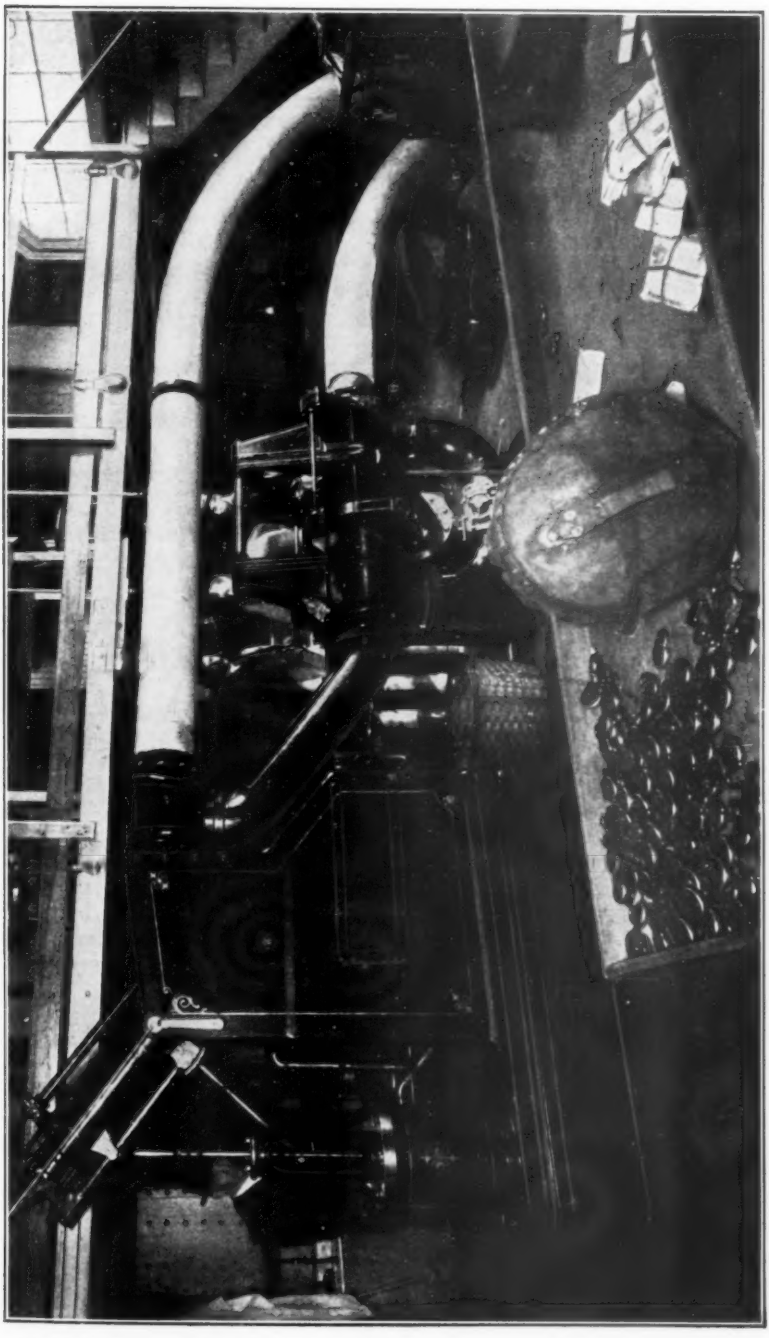

FIG. 133.—PNEUMATIC-TUBE SENDING AND RECEIVING APPARATUS, NORTH UNION RAILWAY STATION, BOSTON, MASS.

Pennsylvania station, where the tubes terminate in the mail room, on the train floor. The distance from the postoffice to the Pennsylvania station is 4,500 feet, requiring 9,000 feet of tube. The carriers cover this distance in one minute and twenty-five seconds. At the Reading Terminal the sending and receiving apparatus is located in the mail room underneath the train shed. The power plant for operating this circuit is located in the power house of the Pennsylvania Railroad at Sixteenth and Filbert Streets. Two duplex Ingersoll-Sergeant air compressors are installed, but only one is used at a time, the other being kept as a reserve. These compressors consist of two Corliss engines with air cylinders in rear and in line with the steam cylinders. (Fig. 123.) The air pressure used is 6 pounds per square inch, and the quantity of air compressed is about 1,200 cubic feet per minute, the compressor running at a speed of 75 revolutions per minute. The pressure varies with the number of carriers in the tube at any given time.

The terminal apparatus used at the Pennsylvania station consists of a sender and open receiver, at the General Postoffice a sender and closed receiver, and at the Reading Terminal two intermediate-station senders and receivers. (Fig. 132.) A set of cut-out switches are located in a vault under Filbert Street, which are used to cut out the Reading station from the main line whenever it may be desirable.

In Boston one line of tubes has been constructed from the General Postoffice to the North Union Railway station, by way of Devonshire Street, Spring Lane, Washington, School, Beacon, Bowdoin, Derne, Temple, Staniford, and Causeway Streets. The distance between the stations is about 4,400 feet. Two 8-inch tubes are used, and are operated from the power plant located in the basement of the General Postoffice. Two air compressors are installed, one being kept as a reserve. (Fig. 134.) The air compressors are of the duplex Corliss type built by the Rand Drill Company.

The tubes terminate at the Union Railway Station (Fig. 133) in the mail room under the tower at the southwest corner of the building. In the General Postoffice the sending and receiving apparatus is located on the mailing floor. The time of the transit of the carriers between the stations is about one minute and thirty seconds.

Unusual difficulties were met in laying the tubes in Boston,

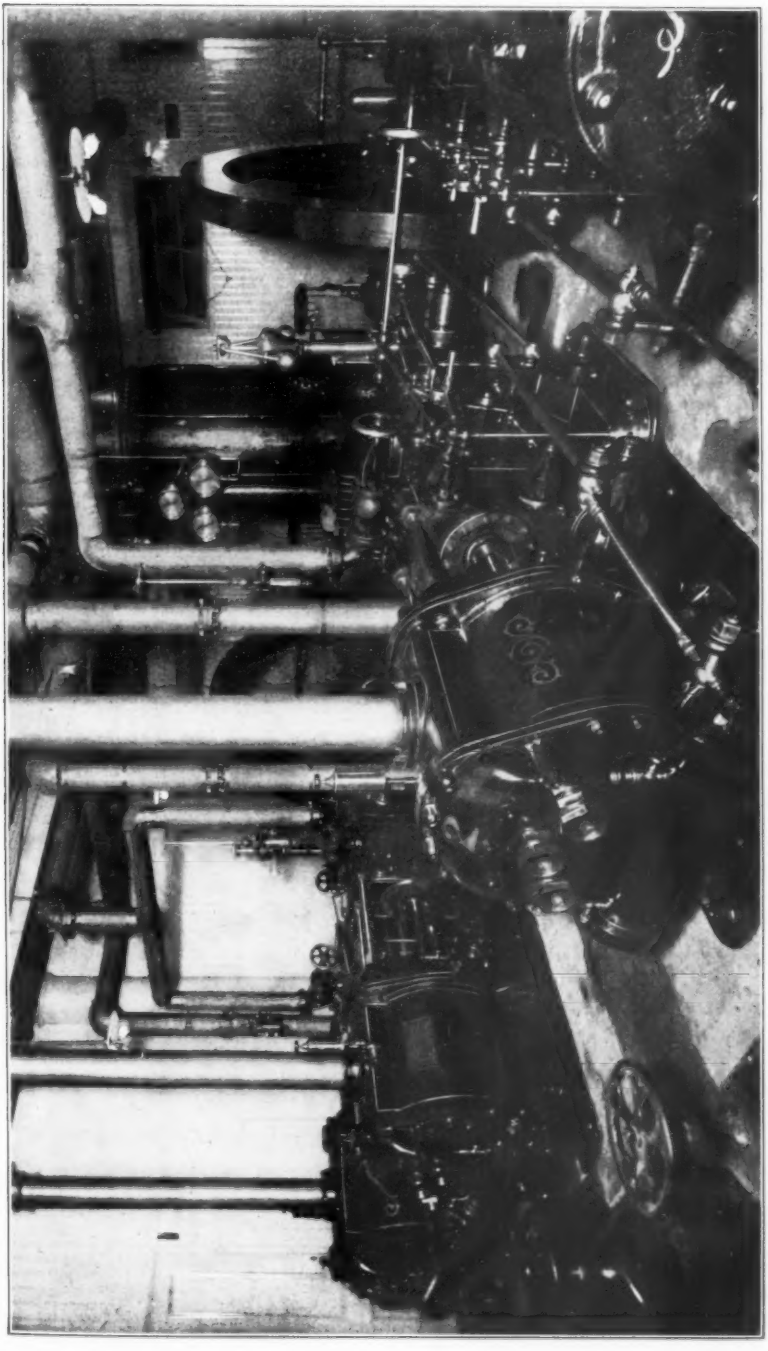

FIG. 134.—AIR COMPRESSORS, GENERAL POSTOFFICE, BOSTON, MASS.

owing to the narrow and crooked streets, which were already crowded with sewers, gas and water pipes, electric conduits, etc. This necessitated laying the tubes deeper in the ground than would otherwise have been necessary.

In New York there are three circuits of 8-inch pneumatic tubes radiating from the General Postoffice.

The first connects the General Postoffice with Branch Postoffice "P" in the Produce Exchange. The tubes are laid along Beekman, William, and Stone Streets, a distance of 3,700 feet. The outgoing and return tubes are operated by a single air compressor located in the General Postoffice, the same as the Philadelphia and Boston circuits.

The second circuit extends from the General Postoffice to Postal Station "H," at Forty-fourth Street and Lexington Avenue, a distance of three and one-half miles, with intermediate stations at Branch Postoffice "D," Third Avenue and Eighth Street, Branch Postoffice "Madison Square," Twenty-third Street and Madison Avenue, and Branch Postoffice "F," Third Avenue and Twenty-eighth Street. The tubes are laid along Center Street, Lafayette Place, and Fourth Avenue, with branches in Twenty-fourth and Twenty-eighth Streets. This circuit is especially interesting, as it is the longest circuit yet constructed, and has the largest number of intermediate stations. The north-bound tube is operated by an air compressor in the postoffice, and the south-bound tube by air compressors located in the power-house at Forty-fourth Street and Depew Place. The initial pressure in both lines is 12 pounds when no carriers are in the tubes. It frequently rises to 17 pounds when the tubes are worked to their full capacity. Carriers are dispatched at intervals of fifteen to eighteen seconds, and it frequently happens that there are fifty or sixty carriers in the line at the same time. The apparatus at the General Postoffice and Branch Postoffice "H" and at Forty-fourth Street consists of a transmitter and open receiver. Two sets of intermediate-station sending and receiving apparatus are installed at Stations "D," "Madison Square," and "F." These intermediate stations can be cut out of the main line by electrically operated switches located in vaults in the street. By means of graduated discs placed on the front end of the carriers mail can be dispatched from the General Postoffice to any desired station on the line, where it will be automatically discharged from the tube. The speed of the carriers in this as

in all the other circuits is about thirty miles per hour, requiring therefor seven minutes for the carrier to go from the General Postoffice to Forty-fourth Street.

The third circuit extends from the General Postoffice in New York to the General Postoffice in Brooklyn by way of the New York and Brooklyn Bridge. The construction of this l ne involved novel features. A large portion of the tubes is in an exposed position on the Bridge (Figs. 136 and 137), subject to

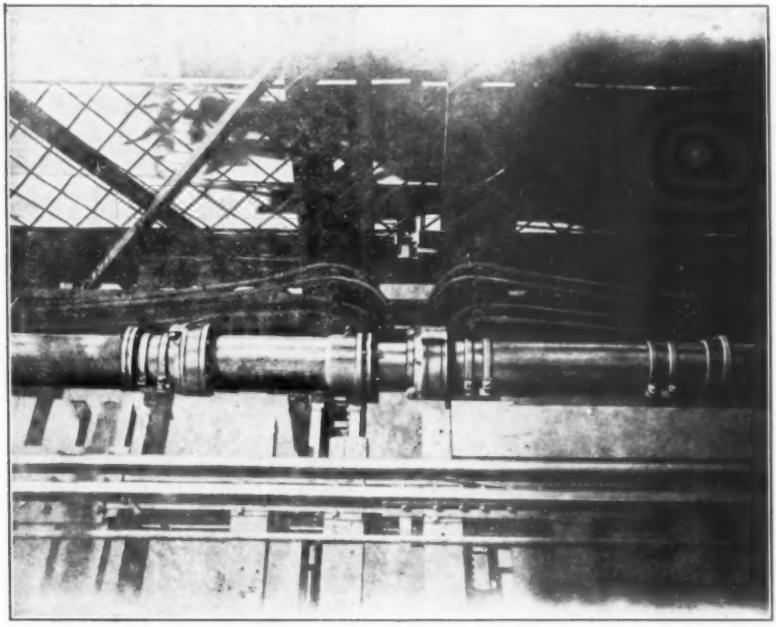

FIG. 135.—EXPANSION JOINT IN THE PNEUMATIC TUBE AT THE CENTER OF THE NEW YORK AND BROOKLYN BRIDGE.

the vibration of that structure and a large amount of expansion and contraction from changes of temperature. When tubes are laid under ground no special provision is made for expansion, but on the Bridge and approaches it was necessary to provide no less than twelve expansion joints, and the maximum motion of these joints at the centre of the Bridge is about 14 inches. The tubes lie along the railway track, supported upon iron brackets clamped fast to the structure. On the approaches to the Bridge the tubes rest on rollers in each bracket, which allows freedom of motion longitudinally.

FIG. 136.—PNEUMATIC TUBES ON THE NEW YORK AND BROOKLYN BRIDGE.

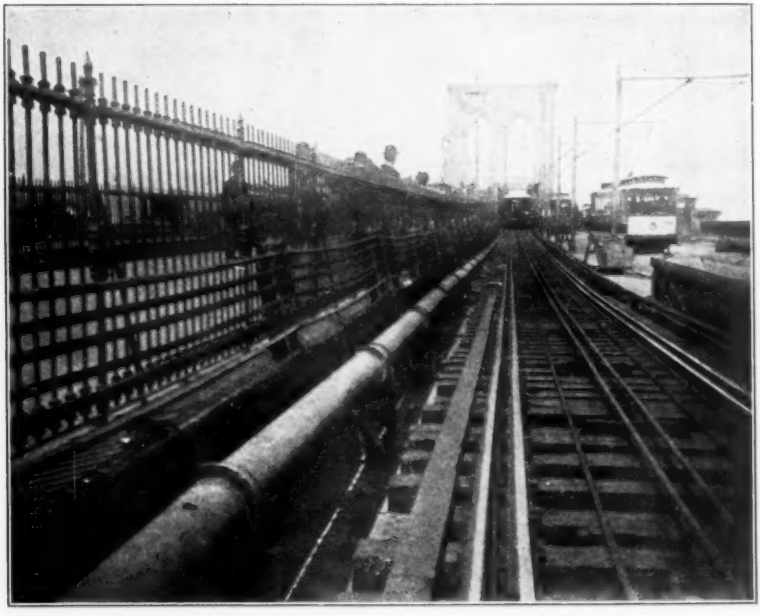

FIG. 137.—PNEUMATIC TUBES ON THE BROOKLYN APPROACH OF THE NEW YORK AND BROOKLYN BRIDGE.

The expansion joints are made of brass tubing sliding through a stuffing box (Fig 135). Ball-and-socket joints are placed in the tube at several points to allow of a slight lateral motion. It was found that the Bridge structure deflects a considerable amount under the load of a passing train, and this made the ball-and-socket joints necessary where the Bridge joins the abutments. The east-bound tube is operated by an air compressor in the New York Postoffice, and the west-bound tube by an air compressor in the Brooklyn Postoffice. The initial pressure at each end is about 6 pounds per square inch, and the time of transit of the carriers between the two postoffices is 3 minutes, the length of the line being about one and three-fourth miles.

The entire power plant in the New York General Postoffice consists of 4 duplex air compressors driven by Corliss engines. One compressor is kept in reserve and can be used, in case of accident, to operate either of the three circuits.

DCCCV.*

METHODS OF TESTING INDICATORS.

BY D. S. JACOBUS, HOBOKEN, N. J.
(Member of the Society.)

THE attention of the writer has been drawn to the fact that there should be some uniform method of testing indicators, through tests which he recently made on a pair of indicators in which the variation of the correct scale from that marked on the spring involved a large sum of money made payable by a contract. These indicators had been tested by using a mercury column as a standard, and the results gave a widely different scale for rising and falling pressures; that is, there were double lines obtained in the test diagrams for each pressure, the lower line being for a rising pressure and the upper line for a falling pressure. On testing these indicators with the testing apparatus in use at the Stevens Institute of Technology there were no double lines traced on the diagrams except for one of the higher pressures with one of the indicators, where the two lines were about one hundredth of an inch apart. The scales of the springs were much more uniform for the various pressures than those obtained by the mercury column, and the difference between the scales obtained by the mercury column and by our method was a large one, and seemed especially so when the amount of premium involved in the contract was considered. In view of the fact that there may be a difference in the results obtained by various laboratories, and also as a standard method will have to be decided on in the report of the committee appointed to codify and standardize the methods of making engine tests, it may be well to describe our methods in detail, and solicit discussion on the same.

The method we have adopted in testing an indicator is to regulate the pressure to some given amount and to trace two lines with the indicator: one after pressing the pencil upward

* Presented at the New York meeting (December, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

and releasing it, and the other after pressing it downward and releasing it. Before taking each one of the lines the indicator is rapped sharply with a small wooden stick in order that the friction shall approach the friction of motion which exists when an indicator is used on an engine. If the indicator is in correct adjustment the two lines should fall one over the other.

In testing an indicator with steam pressure, the steam is first brought to the maximum pressure to which the indicator is to be subjected, and the indicator cock is then opened and closed a number of times in order to heat the indicator. The steam pressure is then released, and the atmospheric line is taken after turning the indicator cock to the proper position. Two lines are traced for the atmospheric line in the manner already described, and one should fall over the other in the same way as for the lines taken at given steam pressures.

Ordinarily for pressures above the atmosphere the lines are taken for increments in pressure of either ten or twenty pounds per square inch, and for springs of a light scale below the atmosphere at intervals of two or three pounds per square inch.

Indicators should be tested both before and after they are used on an engine, or other trial. One object of testing the indicators before the trial is to make certain that there is no excess of friction, which, if present, will form double lines on the test diagrams. If double lines are found it is generally due to some irregular action of the spring in either throwing the piston of the indicator out of line, or in forcing it against the side of the cylinder. All tests, therefore, in which there are widely different lines for increasing and decreasing pressures, are in error either through incorrect adjustment of the indicator, or through some error in the standard used in measuring the pressure. A mercury column will often give incorrect results when used in testing an indicator with steam pressure unless it is allowed to stand a long time at each pressure before taking a reading, on account of the fact that a considerable pressure is required to overcome the inertia of the mercury in the column if the latter is in motion, and there may be pulsations which will affect the result. We at one time employed a mercury column in the form of a glass tube containing the mercury, and with the mercury in plain sight it required the most delicate manipulation to make the column balance at a given mark for a long enough time to obtain correct results. In some

columns the height of mercury is determined by employing electric contact points, and the lines are taken on the indicator diagrams when the top of the column of mercury touches the contact points, no effort being made to bring the column of mercury to a standstill in taking the readings. That there may be a large error in calibration made in the way can be readily shown by running the column at two greatly different speeds, in which case the results will not agree with each other.

It was stated in my paper on "A Comparison of the Mean Effective Pressure of Simultaneous Cards," taken by different indicators, presented at the New York meeting, December, 1893, that there should be no great difference in the lines for rising and falling pressures with indicators in proper adjustment. This fact has been verified by a number of special tests made since then, some of which were published in the "Stevens Indicator" of April, 1894. The fact that the lines should be the same for rising and falling pressures, with an indicator in correct adjustment, when proper precautions are employed as explained later on, may be readily verified by employing the apparatus described herein.

The apparatus used for testing indicators at ordinary pressures above the atmosphere is shown in Fig. 138. The indicator is placed at *A* on top of the cylinder *B*. The cylinder *B* is made of a piece of 6-inch standard pipe about two feet long. The pressure is measured by means of a plug and weight device, *C*, which is spun around so as to eliminate the effect of friction. The bottom of the plug is at the same level as the pipe *D*. The U-shaped pipe *E* is filled with oil. Before starting to calibrate the indicator, the pet-cock *F* is opened slightly, in order to allow any air in the pipe *G* and the siphon *H* to escape. The siphon *H* is surrounded by water contained in the vessel *I*, which condenses the steam which enters it through the pipe *D*, so that when all the air present is allowed to escape through the pet-cock *F*, the pipe *G* and the siphon *H* will be filled with water. *J* is a pet-cock for removing any water that may collect at the bottom of the siphon *E*, after the apparatus has been in use for a long time. The pressure is adjusted by regulating the amount of opening of the valve *K* in the supply pipe *L*, which furnishes steam, water, or compressed air to the apparatus, and also by adjusting the valve *M* in the escape pipe *N*. *O* is a valve for removing any water which may collect in the bottom of the cyl-

inder B when steam is used, and for draining out the water after calibrating an indicator under hydrostatic pressure. The pan of the plug and weight device C is limited in its movement by means of a fork which comes in contact with it only when the pan is in the extreme positions; the two prongs of this fork are shown in section at P and Q. R is a gauge for showing the approximate pressure. The readings of the gauge R are not used in testing the indicator, but as general guide in the use of the apparatus. The diameter of the plug in the plug and weight

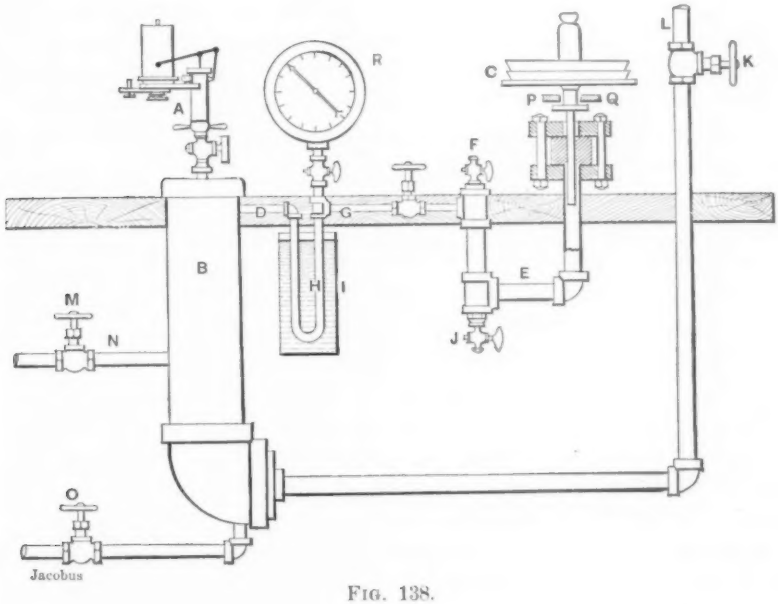

FIG. 138.

device C is 0.5 inch, and the hole in the bushing, 0.5005 inch. Both the plug and bushing are ground true. The average area of the plug and of the hole in the bushing is used in calculating the weight required for a given pressure.

In the apparatus, arranged as just described, one person adjusts the pressure and traces the lines on the indicator diagram. If so desired, one person may adjust the pressures, and a second trace the lines on the diagram. To arrange the apparatus for this purpose, the blow-off pipe from the valve M is led to a point near the valve K, and a second valve, for adjusting the rate of flow through the blow-off pipe, is placed

near the valve *K*, where it can be operated conveniently by the party who adjusts the pressure. The apparatus which we use is arranged in this way.

The apparatus is similar to that described for testing thermometers at temperatures above 212 degrees Fahr. in my discussion of the paper by Messrs. Goubert and Peabody on "Throttling Calorimeters," presented at the New York meeting, December, 1895.*

In testing indicators with steam pressure, the steam is brought to the maximum pressure to which the indicator is to be subjected; the indicator cock is then opened and closed quickly a number of times, in order to heat the indicator. The steam is then released from the cylinder *B*, and the atmospheric line is taken after turning the indicator cock to the proper position. In taking the atmospheric line, as well as the lines for any other pressure, the pointer of the indicator is first pressed upward, and then released and a line taken, then pressed downward and released and a line taken, the indicator being rapped sharply with a small wooden stick before taking each line, as has already been explained. After obtaining the atmospheric line, steam is admitted through the valve *K*, until the pan and weight device is balanced while being rotated. This requires a very fine adjustment, and the line is not taken until there is no tendency for the plug and weight device to either rise or fall.

An apparatus for testing at pressures higher than can be carried on the boilers is represented in Fig. 139. It consists of a pipe *A*, 4 inches in diameter and about 8 feet long, placed in a slanting position, and arranged so that the lower end, from *B* to *C*, can be heated by means of Bunsen burners. The pipe *D* is connected to both a water and steam supply. Water is first admitted through the valve *E*, until it nearly fills the gauge glass *F*, after which steam is admitted, and the water in the pipe *A* is heated, and the pressure, as indicated by the gauge *G*, is brought to that of the entering steam. The valve *E* is then closed, and the pipe *A* is heated by means of the Bunsen burners until the desired pressure is obtained. The pipe *H* leads to a plug and weight device, similar to the one shown in Fig. 138. The indicator is placed at *I*. In the calibration the pressure is brought to slightly above the maximum, and is then

* *Transactions*, vol. xvii., p. 168.

adjusted by opening the valve *J*. The first line taken is for the highest pressure, and immediately after taking the line the cock of the indicator is turned and the atmospheric line is traced on the diagram. The diagram is then removed from the indicator. Several diagrams are taken at the maximum pressure, after which the pressure is diminished for taking the other lines by adjusting the height of the flames of the Bunsen burners, and the amount of opening of the valve *J*. Two lines are taken for each pressure, in the same way as described for the apparatus shown in Fig. 138. *K* is a screen for confining the heat to the

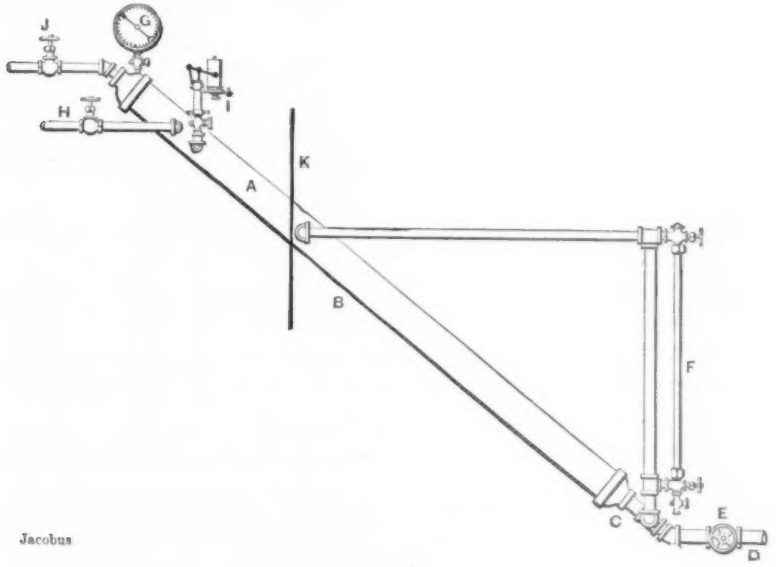

FIG. 139.

lower end of the pipe *A*. The upper end of the pipe *A* is encased with a non-conducting covering, to diminish the radiation. An asbestos shield is placed on top of the lower end of the pipe to partly confine the flames of the Bunsen burners and thus render them more effective.

The apparatus employed in testing springs used under a vacuum is shown in Fig. 140. The indicator is attached at *A* to the top of the horizontal cylinder *E*. The pipe *B* leads to a vacuum chamber and pump, and slopes downward, so that no water can collect in it. The amount of vacuum is regulated by adjusting the valve *C*, leading to the air pump, and by

admitting a small amount of air through the pet-cock *D*. The vacuum gauge at *F* is graduated in inches of mercury and in pounds per square inch, a correction being made in the scale for the amount that the mercury falls in the vessel *S* when it rises in the tube. *G* is a chamber for condensing any steam which may pass through the valve *I*, and thus prevent it from condensing on top of the mercury in the gauge *F*. The water which collects in *G* is drawn off through the pet-cock *H*. The height of the mercury cup *S* is adjusted by means of the screw *U*, so that with no vacuum the surface of the mercury is

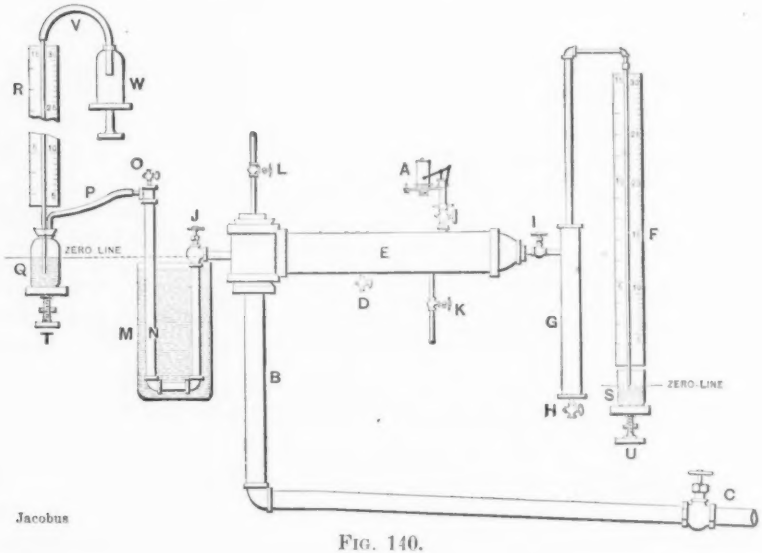

FIG. 140.

on the same level as the zero point of the scale. There is a straight edge in front of the mercury cup *S*, at the same level as the zero of the scale, which makes it easy to adjust the level of the mercury to the proper height.

The apparatus may also be used for testing springs at pressures up to fifteen pounds above the atmosphere, the mercury column *R* being used for this purpose. Where a spring is used for both a slight pressure and a vacuum, the entire calibration can, therefore, be made with the apparatus shown in Fig. 140.

In measuring a pressure above the atmosphere, the valve *I* is closed and the valve *J* opened. The pressure of the steam acts then on the water contained in the U-shaped pipe *N*, and is

transmitted through the hose connection P to the water in the upper part of the closed bottle Q. The lower part of the bottle Q is filled with mercury, and when the pressure increases, this mercury is forced upward in the column R. The height of the bottle Q is adjusted by means of the screw T, so that the mercury is brought opposite the zero line of the scale, a straight edge being used in the front of the bottle, at the same level as the zero line of the scale, as in the case of the vacuum gauge F. In adjusting the bottle Q up and down, the glass tube forming the column moves up and down with the bottle; this tube bears directly against the scale on which the pressures are read. Should the pressure be accidentally raised to such an extent that the mercury will be blown out of the column, the mercury will pass through the rubber hose V into the bottle W.

In testing the indicator, it is first heated by means of steam, admitted through the valve K, until it attains what is considered to be about the average temperature to which it will be subjected in the test. Two lines are taken at each pressure, and all other details in making the tests are the same as have already been described.

It has been already stated that proper precautions must be employed in tests of indicators in order to obtain lines which shall be the same for rising and falling pressures. There may be a difference found in the lines for rising and falling pressures if there is a great difference in the temperature of the parts of the indicator above the piston in taking the two lines, whereas if the temperature is made about the same for each line then one line will fall over the other. Double lines may, therefore, be obtained by starting with a cold indicator and increasing the pressure quickly before taking a line, then raising the pressure to the maximum amount and decreasing it quickly before taking a second line. The difference between the lines taken in this way with an indicator in correct adjustment will vary from about one hundredth to a fiftieth of an inch with a sixty-pound spring. If, however, the pressure is maintained for a short time at a constant figure before taking each of the lines, the lines will fall one over the other. If an indicator is tested with an increasing pressure without heating it before commencing the tests, the variation of the temperature in the spring will be much greater than that which would occur if the indicator were used for taking an indicator card from an engine, and it is to

correct as nearly as possible any error due to this cause that the indicator is brought to about the average temperature in our tests before taking the atmospheric line.

A special test which can be made to show that there should not be a difference in the lines for rising and falling pressures, and to show that the results obtained by our method of testing agree with those in tests where the pressure acting on the indicator is changed quickly over the entire range, as is the case when an indicator is used on an engine, is as follows: The indicator is subjected to a high pressure at the start, and the pressure is lowered, and a line taken at a falling pressure. The indicator cock is then turned quickly so as to reduce the pressure, in the indicator to that of the atmosphere, and is immediately opened again, and a second line taken at the given pressure which will be a line for a rising pressure. As the temperature of the spring and mechanism is not changed the two lines will fall one over the other, thus proving that the same line is obtained for a rising and falling pressure. The atmospheric line is then taken before the indicator has cooled appreciably. The scale determined in this way will be found to be the same as that obtained by employing the method which has been adopted in our tests. In all tests of indicator springs five or more cards are taken, and the average of the ordinates from these cards are employed in calculating the scales.

Table I. gives the results of tests made on an indicator spring by employing our usual method, with the exception that some of the tests were made at a decreasing pressure in order to show that, when proper precautions are taken, the results will be the same as for increasing pressures.

The scale of this spring when the pressure was varied quickly from the atmosphere to 100 pounds per square inch, obtained by the special test which has just been described, was 57.5 pounds, which agrees with the result given in Table I.; the average of four cards which practically agreed with each other was used in computing the scale obtained by the special test.

TABLE I.

TEST OF INDICATOR SPRING, WITH INCREASING AND DECREASING PRESSURES.

Conditions.	No. of Card.	Distance in Inches from Atmospheric Line to Pressures Given Below. Pressures in Pounds per Square Inch.				
		20	40	60	80	100
Pressure increasing	1	.350	.700	1.050	1.395	1.735
	2	.365	.710	1.065	1.405	1.745
	3	.375	.710	1.050	1.405	1.750
Pressure decreasing	1	.360	.705	1.065	1.400	1.730
	2	.355	.710	1.055	1.410	1.750
	3	.355	.695	1.055	1.395	1.730
Average	.....	.360	.705	1.057	1.402	1.740
Scale of spring	.....	55.6	56.7	56.8	57.1	57.5
Distance from 20 lb. line to lines at the higher pressures	.....	.....	.345	.697	1.042	1.380
Scales starting from 20 lb. line*	.....	.....	57.9	57.4	57.6	58.0

*The scales starting from some line higher than the atmosphere are used where back-pressure line of the indicator card is above the atmosphere, as in the high-pressure cylinder of a compound engine. The scales measured from a line above the atmosphere in this way are often more uniform than those starting from the atmospheric line on account of the end pieces, which fit over the indicator spring, becoming worn so that the spring does not bear properly against them until there is a certain amount of pressure acting on the piston of the indicator. This was the case for the indicator spring for which the calibrations are given in the table.

DISCUSSION.

Mr. A. M. Mattice.—The method proposed by the author introduces a very objectionable personal element, viz.: Pressing the pencil up (or down) and releasing it. It needs but a moment's experimenting to prove to anybody that by careful manipulation in this respect widely varying results may be produced. If the pencil be pressed down but a very little way it will not return as far as if it were pressed down farther before being released. Besides, if pressed down a certain amount, found by trial, it will return beyond the point of frictionless equilibrium and remain there. A little manipulation can, therefore, not only show that the indicator is apparently frictionless, but that it has negative friction, if such a term may be used. This is, in fact, shown by the author's Table I. In the column for 20 pounds pressure, the distances from the atmospheric line shown by the three cards for increasing pressure are 0.350, 0.365, and 0.375 inch, respectively, or an average of 0.363 inch;

while for decreasing pressures they are 0.360, 0.355, and 0.355 inch, or an average of 0.357 inch. Here, apparently, the indicator piston did not lag behind on account of frictional resistance, but was actually pushed farther ahead, by some means, than the average. The same anomaly occurs in the 40-pound and 100-pound column, while in the 60-pound column the figures show that the friction acted as a retarding force, as we ordinarily experience it, rather than as an accelerating agent. These differences were probably due to slight differences in manipulation; possibly so slight as to be unnoticed by the operator. A careless operator could, by this method, easily produce very incorrect results; while an expert operator, if inclined to be dishonest, could show the indicator scale to be greater or less, apparently, than the true scale, according as it might suit his interests.

Another objection to the proposed method is the introduction of the wholly artificial element of rapping the instrument to assist the spring in overcoming the frictional resistances. This, undoubtedly, aids in showing what the true value of the spring would be if unhampered by the piston and other mechanism. But what we wish to know for practical purposes is the true scale of the spring under the necessary conditions of actual working. Testing indicators by any method must introduce, to a greater or less extent, some artificial conditions, but it seems to me that the author's method of rapping the indicator will show a greater error in one direction than existing methods show, if any, in the opposite direction. That the indicator piston does lag behind, and shows too high a pressure in falling lines, and *vice versa*, is well known to those who use the indicator. An extreme case, which is often experienced, is where the indicator is so "sticky" as to show a ragged expansion line. Any one who has access to an engine which will show identical cards for an indefinite length of time—such, for instance, as a moderate speed pumping engine working against a constant head—can easily prove the lag of the indicator as follows: Get an indicator which is as nicely adjusted as possible. Now fit to this indicator another piston which will give considerable friction; not so much as to make it "sticky," but within ordinary practical limits. Take several cards with one piston and then with the other, using the same spring in both cases. The piston having the greater friction will invariably show the higher expansion

line. If the indicator were rapped while in use, it would probably show a more correct card, but some automatic arrangement should be furnished for rapping it with sufficient frequency to make the results compare with the rapping while testing at a stationary pressure.

As to the inaccuracy of testing from a mercury column, which the author speaks of, due to the inertia of the mercury, I would say that if it is attempted to bring the mercury column to rest, this trouble will always appear. But when the pressure is constantly increased or decreased at an approximately uniform slow rate, the error is inappreciable. This can be readily done where the electric contact method is used. The errors due to differences of speed of testing with the mercury column are likewise too small to be noticeable unless excessive speeds be used. A very convenient speed is an increase or decrease of 20 pounds pressure per minute. I have never attempted to determine accurately how long it takes for the electric device to draw the record line, but it is less than one-fifth of a second. But, assuming it to be fully one-fifth of a second, with the mercury rising 20 pounds per minute, a 60-pound spring would advance, while the pencil is in contact with the paper, the $\frac{1}{900}$ part of an inch. This is narrower than a fine pencil line, and could not be noticed. Even double this rate, or 40 pounds per minute, would not bring the error within the range of observation. A practically constant rate of rise or fall of the mercury column is easily produced by using a very small steam valve with a stop to limit its opening.

One frequent source of error should be provided against when testing from the mercury column, viz.: When the testing is done slowly, the indicator is not kept at the same temperature as in practical use. This can be easily corrected as follows: Let the indicator have two connections: one direct to the testing apparatus, and a by-pass connection to the source of steam pressure. After the apparatus has drawn a record line on the card, shut the indicator off from the apparatus, connect it to the high-pressure steam, and work it up and down several times, reconnecting it to the apparatus sufficiently before the pressure reaches the next record point.

I fully agree with the author as to the desirability of a greater uniformity in the testing of indicators, but the greatest care should be taken to introduce as few artificial elements as possible,

and to eliminate the personal equation as far as it can be done.

Prof. F. C. Wagner.—There is one point in Professor Jacobus's paper which does not agree with my own experience. He states that in testing an indicator *hot* it should first be subjected to a steam pressure equal to the maximum pressure used. In another place he states that the indicator should be brought to the average temperature to which it is subjected in the test.

A number of years ago I made some tests to determine just what the temperature reached by an indicator spring was when different steam pressures were beneath the piston. The pencil motion and piston rod were removed from an indicator, and with the piston and spring in place, a thermometer was inserted to measure the temperature of the space directly above the piston In every case, no matter what the steam pressure, the temperature was constant and about 212 degrees Fahr.

The reason for this is that the space is filled with steam at atmospheric pressure which has leaked past the indicator piston. The amount so leaking is usually small enough so that any superheat which might be due to throttling is given up to the cylinder walls of the indicator before the steam comes into contact with the spring.

Consequently, it makes no difference what pressure of steam is used to heat up an indicator spring preliminary to a *hot* test, so long as it is greater than atmospheric.

In this connection it is worth while to consider the conditions which obtain when an indicator is used upon the low-pressure cylinder of a condensing engine. During a portion of the time the pressure is above the atmosphere, and during the remainder it is below. There is first a puff of steam against the indicator spring, and then a draught of air from the room as air is sucked into the indicator. The temperature reached by the spring is evidently somewhat between the temperature of the room and 212 degrees Fahr., depending upon the proportionate lengths of time during which a pressure or a vacuum exists under the indicator piston. By actual measurement I have found temperatures varying from 140 degrees to 160 degrees Fahr. in different cases.

When the indicator works entirely upon a vacuum the temperature of the spring may be taken as that of the surrounding air.

Prof. W. F. M. Goss.—An arrangement for testing indicators by means of steam pressure, which has given satisfactory results in the engineering laboratory of Purdue University, is shown by Fig 141. This apparatus has been in use for several years past for checking indicators and thermometers. It consists of a 4-inch steam drum, having a steam supply pipe at one end and a discharge pipe at the other end. At intervals throughout its length are fittings to which indicator cocks and thermometer plugs may be attached, only two of which are shown in the figure. The length of the pipe is such that two or more indicators may be tested at the same time. The steam supply pipe

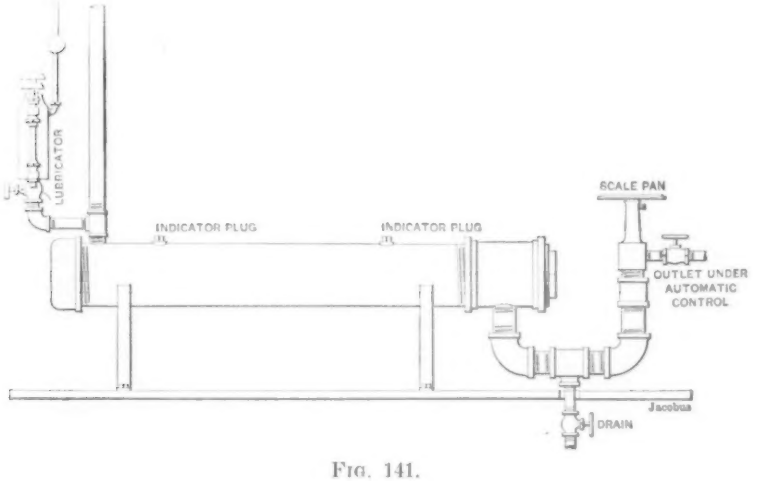

FIG. 141.

has attached to it an ordinary lubricator, and the discharge opening is controlled by an automatic valve which, when set for a given pressure, serves to maintain that pressure within the 4-inch steam drum.

The automatic control will be better understood by reference to Fig. 142. Its principal feature consists of a cylinder and plunger made by the Crosby Steam Gauge and Valve Company, and similar, except as to external dimensions, to the cylinder and plunger employed by that company in their standard "Gauge Tester," and similar, also, to that employed by Professor Jacobus in the apparatus described by him; that is, the pressure within the drum has access to a plunger of known area, which plunger is loaded by the application of weights to a weight

holder. But the apparatus under consideration is different from the others to which reference has been made, in that the plunger not only serves to determine the pressure within the drum, but that it also acts automatically to maintain the same constant. This is accomplished by having the plunger uncover a port, the form of which is such as will give an increase in the area of the port-opening as the plunger rises. The port is small, and the passages leading to the cylinder are large, so that the movement of steam under the plunger does not measurably

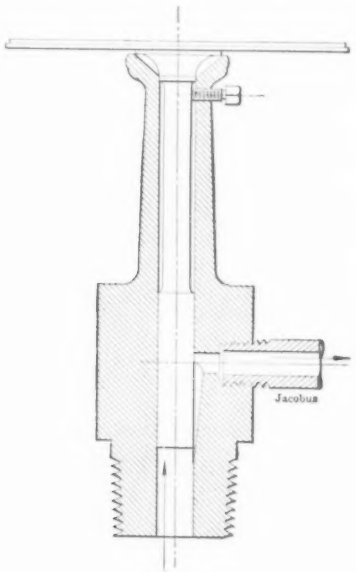

FIG. 142.

affect its pressure. The plunger, therefore, with the load it may carry on the weight holder, can be maintained in balance by the pressure of steam only when the pressure is constant. If an excess of steam is admitted to the steam drum the plunger rises, and in so doing gives a larger outlet. If the supply diminishes, the outlet opening is diminished. The arrangement constitutes a pressure-regulating valve which responds to differences of volume, the pressure remaining at all times constant.

The purpose of the small drain between the plunger cylinder and the steam drum (Fig. 141) is to carry off the water of condensation which, if allowed to pass the automatic valve, gives

rise to unsteadiness in its action. The lubricator on the steam pipe serves to lubricate both the indicator pistons and the plunger of the automatic valve.

In using this apparatus the indicators are attached to the drum, the drum warmed, the small drain (Fig. 141) is slightly opened, weights representing the desired pressure are placed upon the weight holder, the lubricator started, and steam gradually turned on to the supply pipe, the weight pan being rotated meanwhile. When the weight holder floats freely between stops, the indicator cocks are opened and records made similar to those described by Professor Jacobus. Additional weight is then placed upon the weight holder, and the process repeated for a new pressure.

I would add that for testing indicators *cold* the Crosby gauge tester seems to serve every purpose.

*Professor Jacobus.**—The objection raised by Mr. Mattice to the method of pressing the pencil of the indicator up and down and releasing it does not hold, because an indicator in correct adjustment will give the same line after the pencil has been pressed upward as it will after it has been pressed downward, when the indicator is rapped, as described in the paper, irrespective of the amount of pressure exerted on the pencil.

Pressing the pencil up and down and releasing it is simply a test to make certain that the friction of the indicator does not affect the readings, and an indicator should not be used if such irregularities as are described by Mr. Mattice are found to exist.

The scale obtained in the manner described in my paper by pressing the pencil up and down will be the same for an indicator in correct adjustment as the scale obtained by rapping the indicator without pressing the pencil up and down. It is necessary, however, to press the indicator up and down in taking each line even after the indicator has been found to be in correct adjustment, because the friction may increase at any moment through oil being blown from the piston or from other causes, and the results may be affected thereby. The statement that by careful manipulation widely varying results may be obtained applies, therefore, only to indicators which are not in correct adjustment, and indicators in this condition should be

* Author's closure, under the Rules.

adjusted so that constant results are obtained before proceeding to use them.

Mr. Mattice states that there may be errors from pressing the pencil up and down which may make it appear that the indicator has negative friction, and says that this, in fact, is shown by my table giving the results of the calibration of a spring. This cannot occur in an indicator in correct adjustment, as has already been explained, and it is not shown by the table. In taking each line, for which measurements are given in the table for both increasing and decreasing pressures, the pencil was pressed up and down, and two lines were taken, as described in the paper, and these two lines fell directly one over the other. The friction of the indicator does not, therefore, enter the results given in the table, and no argument as to the effect of friction in the indicator can, therefore, be based on the differences found. Furthermore, the differences in the average heights given in the table of the lines at the increasing pressure and those of the lines at decreasing pressures are less than the probable error of the averages themselves; hence, for this reason alone, Mr. Mattice is in error in basing an argument on the differences in the way he has done. The differences of the averages for increasing and decreasing pressures for the heights of the lines at 20, 40, 60, 80, and 100 pounds per square inch above the atmosphere line are .006, .004, .003, .000, and .006 inch, respectively. As the readings are taken to the nearest .005 inch, there may be differences of this amount between the averages even if all the measurements are accurately taken, and there is no accidental variation in the lines themselves. This amount, or .005 inch, is about equal to the maximum difference between the averages. As there may be accidental variation in the lines themselves, due to the differences in the amount that the parts of the indicator are heated, etc., there may be greater differences than .005 inch with all the measurements accurately taken. This makes it still more apparent that the differences between the readings for increasing and decreasing pressures found from the table are too small to form the basis of an argument.

Another objection raised by Mr. Mattice is the rapping of the indicator in testing the spring, in order to reduce the friction, on the ground that what we wish to know for practical purposes is the true scale of the spring under the natural conditions of working. I have made tests to show that the true scale of

the spring under working conditions will be given by the method described in the paper, in which the indicator is rapped in order to reduce the friction. My experiments also show that with an indicator in correct adjustment, the indicator piston does not lag behind to any appreciable extent and show too high a pressure in falling lines, and *vice versa*. Mr. Mattice states that the piston having the greatest friction will invariably have the higher expansion line, and describes an experiment to bear him out. No such variations in the heights of the expansion line can be found with indicators in correct adjustment which will meet the requirements, as to friction, viz.: that there shall be no double lines on the test diagram when obtained, as described in my paper. The results of my experiments to prove this fact are given in Tables A and B. In these experiments the indicator cards were taken simultaneously with two indicators, which are designated *A* and *B* in the tables. The adjustment of the indicator *A* was the same in taking all of the cards. Three pistons of different sizes were used in the indicator *B*. One of these was made to fit as tightly as possible without causing double lines on the test diagram. This piston was a much better fit than the pistons usually furnished with new indicators. The second piston used was slightly less than one one-thousandth of an inch smaller than the first piston, and the third piston was made three one-thousandths of an inch smaller. These pistons are designated as "Tight," "Medium," and "Loose" in the table. An extra set of holes was drilled through the casing of the indicator in order to allow the steam which leaked by the "Loose" piston to escape from the indicator without producing an appreciable back pressure on top of the piston.

Table A gives the mean heights of the expansion lines of the cards measured above the atmospheric line and the ratios of the average heights given by the indicator *B* to the heights given by the indicator *A*. With the "Tight" piston used in the indicator *B* the ratio was 1.10; with the "Medium" piston, 1.11, and with the "Loose" piston, 1.10. The constancy of these ratios proves the fact that there were no differences produced in the expansion line by altering the size of the pistons in the indicator *B*.

The indicator cards were obtained from an engine in which the cut-off was about one-fourth stroke and the initial steam

TABLE A.

MEAN HEIGHTS AND PRESSURES DURING EXPANSION, FROM INDICATOR CARDS TAKEN SIMULTANEOUSLY WITH TWO INDICATORS DESIGNATED AS *A* AND *B* IN THE TABLE.

Mean heights measured above the atmospheric lines of the indicator cards to the expansion lines. The length of the portion measured in each card was 72 per cent. of the length of the card. The indicator *A* was in the same adjustment in all the tests. A "Tight," "Medium," and a "Loose," piston were used in the indicator *B*, the "Medium" and "Loose" pistons being .001 and .003 inch, respectively, smaller in diameter than the "Tight" piston.

NUMBER OF SET OF CARDS.	MEAN HEIGHTS IN INCHES.					
	CARDS TAKEN SIMULTANEOUSLY WITH "TIGHT" PISTON IN *B*.		CARDS TAKEN SIMULTANEOUSLY WITH "MEDIUM" PISTON IN *B*.		CARDS TAKEN SIMULTANEOUSLY WITH "LOOSE" PISTON IN *B*.	
	Indicator *A*.	Indicator *B*.	Indicator *A*.	Indicator *B*.	Indicator *A*.	Indicator *B*.
1	.452	.491	.472	.525	.481	.508
2	.450	.499	.489	.532	.465	.510
3	.442	.496	.479	.534	.450	.486
4	.443	.499	.476	.525	.446	.504
5	.459	.501	.458	.517	.446	.510
6	.479	.521	.475	.543	.453	.500
Average	.454	.501	.475	.529	.457	.503
Ratio of mean heights	1.10		1.11		1.10	

Scale of spring used in Indicator *A* corrected for all variations. 59.3
Scale of spring used in Indicator *B* with the "Tight" and "Medium" pistons..... 53.9
Scale of the same spring when used in Indicator *B* with "Loose" piston..... 54.3

TABLE B.

MEAN FORWARD PRESSURES DURING EXPANSION AND MEAN EFFECTIVE PRESSURES FOR ENTIRE CARDS FROM INDICATOR CARDS TAKEN SIMULTANEOUSLY.

PISTON USED IN INDICATOR *B*.	MEAN FORWARD PRESSURES DURING EXPANSION IN POUNDS PER SQUARE INCH.			MEAN EFFECTIVE PRESSURES FOR ENTIRE AREA OF INDICATOR CARDS IN POUNDS PER SQUARE INCH.		
	Indicator *A*.	Indicator *B*.	Difference.	Indicator *A*.	Indicator *B*.	Difference.
Tight	26.9	27.0	+.1	32.6	32.6	.0
Medium	28.2	28.5	+.3	32.4	32.7	+.3
Loose	27.1	27.3	+.2	33.4	32.9	−.5

pressure about 80 pounds per square inch above the atmosphere. To obtain the mean height of the expansion line the ordinate which is marked *A B* in Fig. 143 was laid off through a point in the atmospheric line 28 per cent. of the length of the card from the admission end. The area *A B C D* between the expansion line and the atmospheric line was then measured, and the mean height corresponding to this area was obtained.

The indicator springs were carefully tested and the equivalent scales determined by laying off horizontal lines on one of the indicator cards taken by each indicator at the height of the lines on the test diagrams. The scale corresponding to each area included between the test lines was multiplied by the area, and

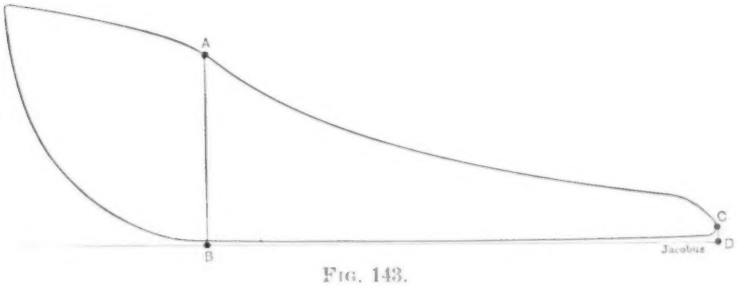

FIG. 143.

the sum of these multiples was divided by the entire area of the card in order to obtain the equivalent scale.

The mean forward pressures during expansion obtained by multiplying the heights given in Table A by the equivalent scales are given in Table B. The mean effective pressure for the entire cards was also worked out, and the results are also given in Table B.

The fact that the mean forward pressure during expansion and the mean effective pressure were the same for the "Tight" piston as for the other two pistons shows that the proper scale of the spring was used for the working conditions. The scale used, as already explained, was that obtained by rapping the indicator so that no double lines were formed on the test diagrams. When the indicator with the "Tight" piston was not rapped double lines were formed, and the scale from such double lines would make the mean effective pressure lower than the true value. The tests prove, therefore, that the correct scale was given by the "Tight" piston when the indicator was

rapped, and that a scale obtained from a test diagram in which there were double lines, using the upper lines for the falling pressures and the lower lines for the rising pressures, would have been in error.

The main error which may occur in using a mercury column is not due to the lack of quickness with which the recording device acts, as might be inferred from the computations given by Mr. Mattice, which show the quickness of such action. The error occurs mainly from pulsations in the long column of mercury, which has a considerable mass and is balanced against an elastic medium. I do not pretend to say that correct results cannot be obtained with mercury columns, but that there can be errors I know from personal experience. In one case we had some indicators tested in the regular way by means of a mercury column in the hands of an expert operator. After completing the tests in the regular way, I suggested that the column be run very slowly. This was done, and the results obtained with the slowly moving column were different from those obtained when the column was run at the ordinary speed. In these tests there were double lines traced on the diagrams in the tests with the mercury column, whereas my own tests gave single lines. The average of the two lines for the slowly moving column agreed with my own calibrations, whereas the average with the column run at the ordinary speed did not. How to make use of the double lines on the diagram is an open question. I know of a concern employing a mercury column which gives only the readings for increasing pressures in their reports, unless the parties for whom the tests are made request that they shall be furnished with the results for decreasing pressures as well as those for increasing pressures.

A point is raised by Mr. Mattice and Professor Wagner in regard to the temperature of the spring being different in the tests to calibrate it than when the indicator is used on an engine. Professor Wagner is correct in saying that the temperature of the spring for the usual working conditions of an indicator for pressure above the atmosphere is about 212 degrees Fahr. I have made tests to determine the temperature to which the springs are subjected when calibrated on our apparatus, and find that there may be superheating in the steam above the piston of from 10 to 20 degrees Fahr. when the indicator is subjected to steam at 80 pounds pressure. The steam which enters our

apparatus contains considerable moisture, and water appears at all times at the drip valve marked *O* in Fig. 138, and should this be closed it raises in a gauge glass which is not shown in the figure. The steam acting on the piston is, therefore, always in a moist state. We have made tests in which a small amount of water was blown into the indicator along with the steam. To accomplish this the blow off pipe *N* was made to lead from the cylinder *B* at the same height as the pipe leading to the indicator, and the drip valve *O* was closed so that water collected in the cylinder. When this was done, the temperature above the piston was found to be 212 degrees Fahr. Tests were made by calibrating an indicator spring at 212 degrees by allowing the water to leak past the piston along with the steam, and by calibrating the same spring in the ordinary way. The difference in the results of the calibrations was less than one-half of 1 per cent., the scale with the spring at 212 degrees Fahr. being slightly the greater. We have considered this difference an unimportant one, and have employed the ordinary method in our calibrations.

In some tests we have made we have used the apparatus shown in Fig. 138 in the same way as the apparatus described for high pressures shown in Fig. 139; that is, we maintain the steam in the apparatus at a given pressure and take five lines or more, together with the corresponding atmospheric lines, before taking lines at another pressure. When this is done, the lines can be taken quickly on the indicator, and the temperature of the spring will be more nearly that which exists under working conditions. The atmospheric lines are taken after tracing the lines at the given pressures, two three-way cocks being connected between the indicator and the cylinder *B*, and both being turned in taking an atmospheric line, so that any leakage in the indicator cock will not affect the atmospheric line.

In the case of a spring working under a vacuum we strive to make the temperature of the spring, in the calibrations, as nearly as possible that to which it was subjected when in use.

The arrangement described by Professor Goss provides an easy and accurate way of regulating the pressure, as well as an accurate means of measuring it. For great refinement, however, I feel that the device which we use, in which the rotating plug is maintained at a constant temperature, will give more accurate

results than where it is heated by the steam. The expansion of the plug and the bushing on being heated by the steam produces an error of about 0.3 of a pound per square inch at 80 pounds pressure. We have tested a plug arranged as described by Professor Goss, and have found that there was a practical agreement with our own when the steam which escaped through the opening at the side of the plug issued at a considerable velocity. When the pressure was regulated so that little or no steam escaped there was a difference found of about 0.3 pound at a pressure of 80 pounds per square inch, or the difference corresponding to the difference in area of the plug and of the bushing when cold and when heated by the steam. The reason that there was an agreement when considerable steam was allowed to escape through the opening at the side of the plug arranged as described by Professor Goss was on account of the fact that the frictional resistance of the steam, in passing through the opening beneath the plug, caused a slight loss of pressure between the reservoir to which the indicator was attached and the plug, and when the velocity was such that this loss became equal to the difference of pressure corresponding to the difference in the area of the plug when hot and when cold, the two effects counterbalanced each other.

DCCCVI.*

TOPICAL DISCUSSIONS AND NOTES OF EXPERIENCE.

No. 806—139.

Does it pay to pickle ordinary castings?

Mr. Orosco C. Woolson.—It occurred to me when I saw this question of pickling was to come up that I must make a few remarks, prompted from observations and personal experience in this character of work.

To begin with, however, I am impressed with this view of the problem:

1. It depends on several conditions whether it be desirable to subject castings to a bath of pickle or not. For instance, some foundries have a class of work requiring such a soft character of metal, and require such a character of tooling, and, furthermore, have such methods of moulding and pouring, that there is not an adequate return for the outlay in proper pickling; while, on the other hand, there is a class of work necessarily produced in localities and under such conditions of metal moulding and casting that it seems to be advantageous to pickle the castings before subjecting them to the particular character of tooling necessary, so that to my mind it all depends on the several conditions whether it is wise to pickle or not to pickle.

2. If pickling is decided on, for any particular reasons, then I wish to make a suggestion in the interest of the mechanic. See to it carefully that your apparatus is well equipped for thoroughly rinsing before subjecting the castings to the men who are to handle them and do the tooling, for where the rinsing is carelessly done, there will arise sulphurous fumes which are exceedingly disagreeable, and I have no doubt injurious to the lungs of any one, especially if such person is naturally weak in these parts.

There is only one sort of being who can stand it for any great length of time, and that being is "*a once-was*," and he is usually

* Presented at the New York meeting (December, 1898) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

nowadays to be found in museums, done up in medicated sack-cloth.

Mr. Charles L. Newcomb.—I would like to say that the adoption of the sand-blast for cleaning castings replaces pickling in our establishment. The answer to the question asked should depend very largely as to the kind and quality of castings being made. From my experience with the sand-blast in cleaning castings during the last two years, I believe any foundry making over six tons of castings per day cannot afford to use any other method, if the castings must be cleaned. If a foundry is making fifteen to twenty tons per day, there is then more reason for using the sand-blast, because of the saving over any other method of which I have knowledge. There are many foundries in which the sand-blast would not be useful, such as pipe foundries and foundries making car wheels, etc. Foundries using pickle for removing the sand and scale from their castings should immediately join the procession of progress and adopt the sand-blast. Foundries producing steam-engine castings, air compressors, and steam-pump castings cannot afford to be without a sand-blast apparatus for cleaning their castings. Such foundries will find it the only "up-to-date" method of removing the core sand from the steam ports and jackets of all kinds of cylinders; it has no competitor in any form of pickle, either in cost or quality of work. Ports and jackets can be made absolutely clean; it does away with the slow methods usually in vogue where steam cylinders are to be cleaned, which is two or three days of pickling, with hours of labor by some workmen scouring through the ports with a piece of sheet metal. There is sure to be sand in the corners after a casting comes from the pickle, no matter how long it is left on the pickle bench or how thoroughly the pickle is applied. Time is another important element. A plain cylinder 20 inches in diameter of bore and 29½ inches long can be cleaned by one man in a properly designed sand-blast room in ten minutes, and the ports will be cleaned. Sand-blasting a casting removes much of the silicious oxide from the casting, so that the parts which are to be tooled do not destroy the tools so rapidly; especially is this noticeable where castings go to the turret and milling machines, and the tools which are used are expensive, and a small saving is worth consideration. Good, honest jobbing foundries should hail the use of the sand-blast, because it shows up all the imperfections in the castings, cleaning all sand out of the blow-holes and stoggy

places, enabling the imperfect casting to be discarded in the inspection. One operator in the operating room can clean fifteen tons of steam-pump castings per day, of ten hours, requiring a No. 95 B. O. & R. A. Tilghman mixer (sold by Ward and Nash; Boston); a boiler plate washer, 7 feet long, 4 feet diameter; a Sturtevant exhaust fan, 40 inches; Deane air compressor, 10 by 12. The usual pressure being between 15 to 22 pounds, using a nozzle of $\frac{5}{8}$ inch in diameter.

Mr. Suplee.—I understand that in Germany they are now using a weak solution of hydrofluoric acid with great success. It acts on the silica, and eats out the sand in a way that an ordinary pickle does not.

The solution used for this purpose is composed of about twenty parts of water to one part of a 30 per cent. solution of hydrofluoric acid, which will attack the silica direct, without having any perceptible action on the iron. It should be used in lead-lined tanks, and handled with care, as it will injure the hands unless they are protected by rubber gloves. As the ordinary pickle can only act by undercutting the scale and eating away the metal until the scale is removed, it should follow that a solution which acts directly upon the silica is more desirable.

Mr. Wm. Kent.—I have seen the use of hydrofluoric acid for removing the clay from castings that have been cast in baked clay moulds, and it did that very successfully. The hydrofluoric acid seems to have a field for some purposes.

Mr. Newcomb.—I think I ought to say that the sand-blast takes off the silicious oxide on the outside of the casting, and the men in the shop notice it right away. In that way the tools retain their edge.

Mr. Suplee.—In regard to the sand-blast, I know that it is used with remarkable success in preparing surfaces for the Bower-Barff process, and it is practically impossible to get a continuous coating of magnetic oxide unless some such method of cleaning is employed. This leaves the iron with a silvery white appearance, practically chemically clean. This method is used for small hardware and castings of that sort.

No. 806—140.

What is the strength of pipe fittings made by a casting process?

Mr. H. H. Suplee.—I might state that that question is introduced by me at the request of a gentleman not a member of the

Society, for a specific reason. At the present time American manufacturers are exporting large quantities of American wrought-iron pipe for steam and gas and other work of that sort. But they find themselves seriously hampered in meeting the foreign market for pipe fittings. That is, they cannot take the whole order, simply because there is a general mistrust abroad of cast fittings. Wrought fittings are the rule instead of cast fittings, the idea being that the American cast fittings are not strong enough. Under those circumstances it is very desirable to be able to show the foreign users of American products that the cast fitting is all right when properly made, and it was thought desirable to obtain some expression of opinon on the part of a responsible body like this that would boost up the American article, on the other side of the ocean. If anything could be said on that subject it would be of value to American commerce abroad.

Mr. Wm. Kent.—Two cases in the last two years were called to my attention of the breaking of cast-iron fittings; but they were caused by water ram in a steam pipe. Probably if there had not been any cast-iron fittings there, something else would have broken, such as the steam chest of the engine. The cast-iron fitting is apt to be not strong enough to stand a water ram.

No. 806—141.

What constitutes a seamless tube?

Mr. Henry Souther.—A topic for discussion has been placed before the Society which, under ordinary circumstances, should hardly require discussion—"What constitutes a seamless tube?" It is a fact, however, that during the last few months this name has been applied by men whose opinions are of great value in the manufacturing and scientific world, to material which should not be put under this head. It is doubtful if there is authority in the shape of a dictionary or encyclopædia which will settle this matter finally, and we may be obliged to rely upon custom and metallurgy to say whether a tube is seamless or not.

Assuming that the question is one of terminology, let us see what the dictionaries say. Under "seamless" we find in all cases as follows: "Without a seam." What are the definitions for "seam"? Consulting Webster and the Century, it is found that each gives several definitions, referring particularly to cloth, leather, and other fabrics which have no reference to the present

case. Another is also given which seems to hit the case under discussion. It is

"Seam, joint, suture, line of union."

From this definition it appears that no interruption of material need exist or be seen in the finished product. Therefore, relying solely upon terminology, a butt or lap-welded tube, or a tube jointed in any manner, cannot be properly called seamless.

Custom points out very plainly exactly what is meant by a seamless tube. For many years we have been making tubing by various processes of rolling a strip about a mandrel and welding the edges of the strip together. Up to the last few months I have never heard any one call material so made, seamless. The temptation is great to apply the name of a superior article to one that is inferior, for trade purposes, but for the sake of accuracy, if for no higher motive, we should confine ourselves to the distinction and meaning as applied by common usage to the word "seamless."

Examine the process of manufacture of the two kinds of tubing which are recognized as being seamless or jointed. To make either an ingot is cast. The ingot is then rolled to a bloom, the bloom to some form of billet or slab, and this, to form the welded tube, is rolled into what is known as "skelp." So far this steel is seamless. The continuity of the metal has not been interrupted at any point except to cut the steel into pieces small enough to handle. From this material known as "skelp" either a seamless or welded tube can be made, and right here occur those operations which draw a sharp line between the two forms of tube. To make a seamless tube this skelp would be cut into pieces perhaps 10 inches square; then by successive cold-cupping operations, which in no way separate the grain, this sheet of skelp is made to assume the form of a deep, narrow cup, very much of the proportions of a cartridge shell. The closed end of this cup is cut off, and the product is unquestionably a seamless tube, ready to be elongated to suit requirements. This tube has never had the particles of metal which form its sides separated in any way.

From the skelp, to make a welded tube, the operation is entirely different. The long strips of steel are heated in a furnace, passed through a die and over a mandrel in such a manner as to roll it up so that its edges nearly touch. This rolled strip is then heated again to a welding heat and again passed through a die and over

a mandrel, with sufficient pressure between them to cause the edges to unite and make a weld. This weld may be absolutely perfect and it may not. Even if perfect, the particles of steel forming the walls of the tube have been separated and again united by combined heat and pressure. Surely no one would

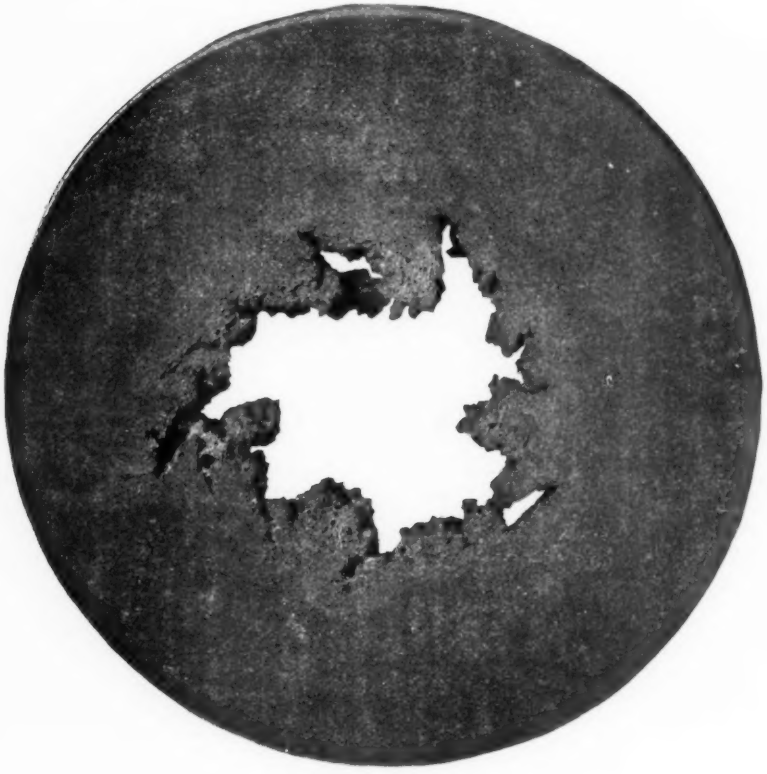

FIG. 144.

claim that steel is united under any conditions as well as it is when melted.

The walls of the seamless tube were united when melted. The walls of the welded tube require combined pressure and heat, and the junction is acknowledged to be more or less imperfect. The most perfect union is visible when examined under the microscope under conditions similar to those used to study the microstructure of metal. It would seem from this that a tube cannot be called seamless because the joint is not visible to the naked

eye. Some joints united by brazing are exceedingly difficult to locate, but the tube is nevertheless an imperfectly jointed one and certainly not seamless.

Still another way is employed to make a seamless tube. The ingot is rolled into the bloom, and the bloom into a round, perhaps 3 inches in diameter. This round is then cut into short

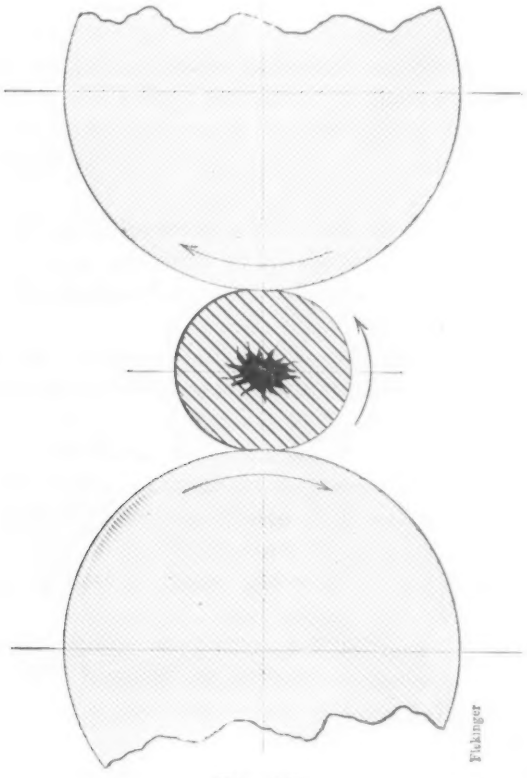

FIG. 145.

lengths, say 12 inches to 15 inches. Through the centre of this round, a hole is then formed, perhaps by drilling cold, perhaps by piercing hot, by one of various methods. In any case, the hole is put through the billet without interrupting the contact between the particles of the wall of the tube. The piece is then a seamless tube, ready to be drawn to any desired size.

The custom-house makes a distinction in import duties between tubing made by one of these processes and another, calling

one seamless and another welded or jointed. The custom-house is rather a good judge of the proper name for things. It therefore seems to me that common usage and process of manufacture draw the line very sharply between a seamless and a jointed tube.

Mr. P. J. Fickinger.—The query, "What constitutes a seamless tube?" is of interest to me in particular. The first thought was that it would be a tube made from a solid billet. The conclusion arrived at by my way of reasoning was that the process of producing a tube was not what was to be considered; it was a question of results only. For if the results are not satisfactory, and a tube after being formed shows any lineal defects when tested to destruction, I cannot see how it can be called a "seamless tube."

In piercing a billet by the rolling process there is quite a tendency to form longitudinal cracks or seams, as you will see by the sample of a section of a bar cut from a 6-inch round (Fig. 144). This bar is of the regulation railroad axle material, and was rolled with a two-roll device as shown in Fig. 145, the axis of the rolls being in parallel with the axis of the bar and in the same plane.

Now, is it not plain to be seen that the principle is the same as in the billet-piercing machines which accomplish the results by the rolling motions, the only conditions lacking being the piercing mandrel, and the axis of the rolls being changed sufficiently to give to the billet or bar the required feed or longitudinal direction?

Now, this is the question I would like to ask this Society: Is it not reasonable to suppose that in the method which has been described, providing all the conditions are not of the best, a seamed tube will be the result?

It is a well-known fact among mechanics familiar with the hot forging of metals, that what is technically known as a "cold shut" cannot be welded and must be cut out.

I wish to call attention to the formation of the cinder which is shown in the cracks of the specimen submitted. Perhaps some of the members present can explain this.

To come back to the subject, "What constitutes a seamless tube?" it appears to me that the best way to dispose of this is to produce the tubes, test them to destruction, and, providing no seam or lack of uniformity is discovered, it is seamless.

If it shows any of these defects it is not, regardless of the method of manufacture.

No. 806—142.

On how small a tool does it pay to put an individual electric motor?

Prof. Dugald C. Jackson.—As a general rule, the following statement, quoted from page 1057 of vol. 18 of the *Transactions*, fairly represents the answer to this query. It is very seldom that it pays to subdivide the power to a greater degree than is here indicated. "All large tools or machines, such as use from five to seven and one-half horse-power and over, should be supplied with individual motors, while smaller tools or machines requiring less power should be grouped and driven from motor-driven shafts. These groups should ordinarily be arranged so that a motor of not less than from three to five horse-power capacity is required, and not more than from ten to fifteen horse-power. The grouping of tools, the subdivision of power, and the manner of delivering power of motors to driven machinery, it may here be said, is a matter which can be given only the most general treatment as a whole, as each industry includes conditions of its own which must be taken into the count. Observation indicates that some manufacturers who are using electrical power failed to weigh carefully the question of its subdivision when preparing to install their plant, and have thereby lost much of the advantage which may be derived from the electrical transmission."

Mr. Jesse M. Smith.—This seems to me to be a commercial question pure and simple—how much does it cost to put the motor on to the tool? Certainly after the motor is there it costs less power to run it by electricity than by belts. But whether for any particular tool it will pay to have an electric motor put on it, is simply a matter of dollars and cents, which must be determined, not only by the tool builder, but by the purchaser of the tool. The advantages of electric power in driving tools or machines of any kind are certainly being recognized more and more every day, and motors are getting cheaper and cheaper every day, so that after awhile the machine tool builders will build their own motors. Then we can get motors at reasonable prices, and then we will have the ideal transmission.

Mr. Halsey.—While not looking at the matter over the standpoint contemplated in the query, it may be worth while to say that at the Baldwin Locomotive Works, where machinery

occupying acres and acres of floor space is driven by electric motors, they have a rule which forbids the use of motors of less than five horse-power under any circumstances. This being the smallest unit of power, it would naturally follow in most cases that small tools should be arranged in groups to consume in the aggregate at least this amount of power. The reason for the rule lies in the fact that, considered as machines, small motors are delicate in construction, and require a disproportionate amount of attention and repairs. It should always be remembered that the individual motor system involves a large number of small delicate motors to keep up, while the group system involves a small number of large and substantial machines.

No. 806—143.

Have you any new notions on machine-shop floors?

Mr. Charles T. Newcomb.—I do not know that I have any new notions, but my practice has been for the last seventeen years, for ground floors in a machine shop building steam pumps or like machinery, to use, for joist or floor timbers, 6 inches by 6 inches kyanized timbers placed 4 feet on centres, well bedded in place, and laid crosswise of shop so as to put plank lengthwise of shop. On top of the timber I put 3-inch matched spruce plank, planed on one side. Before the planks are laid the earth fillings under plank are well puddled for about 2 feet and left higher than floor timber, and struck off to height of floor timbers, leaving a solid surface under plank onto which machinery can be set without other foundations. It is my experience that this kind of floor will wear out on top in about five years, when it will be found to be rotted on bottom so as to need replacing, the kyanized timber lasting indefinitely.

Kyanized timber or treated lumber can now be obtained very readily on the market. This kind of floor costs laid 11 cents per square foot, and is easily replaced, and in view of the fact that the top of floor wears out, it is in my opinion better to use this or similar cheap construction and have a new floor when needed.

For second-story floors for such work as is usually put on such floors, I use 3-inch matched spruce plank with paper laid over it to keep the dust and dirt from passing through, the top flooring being $1\frac{1}{8}$ inches thick, matched hard maple, bored and blind nailed, not wider than $3\frac{1}{2}$ inches. This kind of a floor will last many

years, wearing out in spots only, for instance, where the chips in front of drill or planer are, and upon which the operator stands. The spots can be easily and cheaply patched. This kind of a floor costs 14 cents per square foot laid.

Mr. Albert A. Cary.—I don't know whether I have very much to say in the way of new notions about machine-shop floors, but I have experimented with floorings in manufactories. I had a good deal of trouble where there is a great deal of trucking of heavy machinery and rather heavy, bulky articles. There the floors would wear out very rapidly. So I made up my mind to determine which was the best floor, and I laid a series of sections, starting with spruce, then putting in yellow pine, then taking a better quality of yellow pine, which I believe is known as "cone grey" pine, and afterwards maple, so that we had a chance to test them. I have forgotten the relative length of time that the different floors wore, but the final result was that I ripped up all other kinds of top-flooring course, and laid the maple floor, and I do not know anything that will wear any better than that. One thing convinced me of its splendid adaptability to a place of that kind where there is a good deal of trucking. I found that a number of piano warehouses had adopted it in this city. They roll heavy pianos with small casters constantly over the floor, and that was the one thing they found that would not grind under rollers and would stand. Of course, the poorer qualities of maple will grind under casters, but if you get a good quality of rock maple you cannot get any better floor for a machine shop, I think.

Mr. Durfee.—In one of the machine shops erected by me the lower floor we built in this way. We covered the surface of the ground, after it had been excavated to a proper depth, with 10 inches of broken stone. On that broken stone we bedded some sills of chestnut, and then filled the space between them with broken stone. Those sills were flattened on their upper surfaces, and on them were laid 4-inch planks about 5 inches wide. Those planks were placed with a slight open joint, not to exceed an eighth of an inch, sufficient to allow air to pass down through the cracks and to allow water thrown upon the floor to pass away. Those planks were laid at an angle of 45 degrees with the traffic. I think that an advantage, because when you are hauling loads through the shop you are running at an angle to the grain of the wood. I do not expect to live long enough to see that floor wear out. The

galleries of that shop were made of spruce plank, 4 inches thick, and in no case over 8 inches wide, grooved for inserted tongues, and laid upon girders 10 feet apart. On the top of these planks was laid a hard-pine floor $1\frac{1}{8}$ inches thick, made of $1\frac{1}{4}$ stuff. It was not tongued and grooved. I don't think tongued and grooved floors are suitable for shop floors. This hard-pine flooring was what was called, in that locality, rip-sawed. There were no outside pieces of the log in the boards. The grain ran perpendicularly through the boards as nearly as possible. The edges were laid close and fastened by nails driven right through the boards, which were laid at an angle of 45 degrees to the plank below. That answered a double purpose. First, it diminished the wear of the floor by placing the grain of the wood at an angle of 45 degrees to the traffic over it; and secondly, it stiffened the building very much, as the galleries formed an immense horizontal truss to resist vibration. I have had a good deal of experience in the floors of factories, and I have always found that where the floor—a top floor—was laid with tongued and grooved boards, the floor was practically all worn out when it was half worn out, so to speak. When the floor was worn down to the tongue, it was full of all sorts of disagreeable inequalities, and had to be repaired very frequently. Then that is another point in laying hard pine, or any other wood, unless you are careful to have the grain of the wood run perpendicular to its surface. In other words, if there are slab grains in the wood, those peel up and separate from the boards below. Then you have inequalities, arising from that cause, which are very disagreeable.

Mr. Cary.—I do not know that I can agree entirely with Mr. Durfee as to the use of non-tongued and grooved flooring for a top floor. I have used that. I have never used anything over 3 inches in width, which was nailed down. When ordered, it was drilled to take wire nails, which were driven in just above the tongue so they would hook up against each other. It is well-seasoned wood, and nothing over 3 inches in width. I have known those floors to last eight or ten years, and they are not worn out now, and they have had some pretty rough usage.

There is another thing, that may be a little out of order here. It is not directly a machine-shop floor, but for the treads of stairs in factories. I saw something in Hartford a number of years ago, and I have since tried it myself. There they take maple and make the tread of wood about, well, I should say an inch or an inch

and a quarter thick, and the ends of the boards are sawed off across the grain, and this is placed so as to stand that grain up on the tread, and that is cut up into little blocks one inch square, and so placed on top of the tread. It is end wood. I know of one flight of stairs that I have laid that way. It has been tramped over for a great many years. It is in splendid condition, and the other stairs are worn away and new treads have to be put in; but that seems to wear for an indefinite length of time.

Mr. Durfee.—In the matter of treads of stairs, I saw in England some years ago stairs covered with thick slabs of lead. The lead was certainly half an inch thick, and there was a solid sensation in treading on those stairs which was very satisfactory indeed, and I was assured that the wear was as satisfactory as the feeling.

Mr. Percy A. Sanguinetti.—I would say, from many years of observation, that a floor composed of maple planks, with the joints laid diagonally with the walls of the building, made the most durable covering that I know of.

A floor so laid, made of pieces not over 6 inches wide and $1\frac{1}{4}$ inches thick, all joints tongued and grooved, showed no injurious wear for about five years.

Mr. W. P. Dallett.—The Pennsylvania Railroad in Philadelphia are making an experiment at the Broad Street station, and they are using on one of its stairs a composite tread of lead and, I think, cast iron; that is, the lead in grooves in the cast iron. It has seemed to be very satisfactory.

PAPERS

OF THE

WASHINGTON MEETING

(XXXIXth)

MAY 9th TO MAY 12th, 1899.

DCCCVII.*

PROCEEDINGS

OF THE

WASHINGTON MEETING

(XXXIXth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

May 9th to 12th, 1899.

LOCAL COMMITTEE.

FRANK B. KING, *Chairman*,	GEORGE W. MELVILLE, *President*,
A. H. RAYNAL,	CHAS. E. FOSTER,
W. E. SCHOENBORN,	WM. HARKNESS,
HERMAN HOLLERITH,	ARTHUR E. JOHNSON,
HERVEY S. KNIGHT,	WALTER ASHFIELD MCFARLAND,
GEO. L. MORTON,	HAROLD P. NORTON,
CHAS. E. ROMMEL,	GEORGE R. SIMPSON,
WM. B. UPTON,	ANTHONY VICTORIN,
WILLIAM WALLACE,	WINTHROP COLE.

The City of Washington was selected by the Council, early in the winter, for the place of the spring meeting of 1899, by reason, first, of the fact that the president of the Society for the current year was the Engineer-in-Chief of the United States Navy, and because at the time this decision was reached the status of the naval engineer had not received the consideration which made the passage of what has been called the Naval Personnel Bill an imperative necessity. The attractions of the city itself, the date selected, and the attractive programme which had been prepared

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* Forming part of Volume XX. of the *Transactions*.

by the efforts of the Local Committee, made the Washington meeting a memorable one in point of numbers and in the pleasant memory which it has left to all those who were privileged to attend it.

The opening session of the convention took the form of a most enjoyable reception in the Corcoran Art Gallery, on New York avenue, E street and Seventeenth street, N. W., upon Tuesday evening, May 9th. The secretary, at the request of the Local Committee, introduced Col. T. A. Bingham, United States Engineers, who, on behalf of the Commission intrusted with the control of the District of Columbia, welcomed the Society to Washington, and referred particularly to some of the interesting features connected with the engineering of the Washington Monument. He referred to its foundation, the work of placing the cap, the elevators, and the observations which had been made upon a plumb line suspended in the centre of the structure. President George W. Melville made a brief response on behalf of the Society, and the informal ceremony of introduction to the President of the Society and Mrs. George Westinghouse proceeded in the upper hall. The Marine Band of thirty-four pieces was located in the main hall, and added greatly to the pleasure and brilliancy of the gathering. The art treasures of the gallery, both in paintings and sculpture, were greatly enjoyed. This reception was permitted to the Society by the trustees of the Corcoran Gallery, while the Society itself and its president assumed the burdens of expenditure connected with the affair.

Second Session. Wednesday Morning, May 10th.

This session was made the opportunity for business connected with the convention. The meetings for the reading of papers were held in the large assembly hall of the Arlington Hotel, Vermont avenue and Fourteenth street, and the headquarters of the Society were provided in the adjoining parlor, which made an arrangement of particular convenience, and which gave great satisfaction.

The session was called to order at 10:30, with President Melville in the chair. The plan of registration, which had worked so well in previous conventions, was again carried out, and seven reprints from the register were issued during the convention. It will appear that the numbers in attendance were exceptionally

large for a spring meeting. The registration showed the following members in attendance :

Albree, C. B.
Aldrich, Wm. S.
Allison, Robert
Almond, Thos. R.
Ames, Wm. L.
Andrews, W. J.
Archer, E. R.
Ashley, F. M.
Ashworth, Daniel
Baldwin, Stephen W.
Baldwin, Wm. J.
Ball, B. C.
Bancroft, J. S.
Barnes, A. T.
Barnes, W. F.
Basford, G. M.
Bates, A. H.
Bates, Edw. C.
Bates, E. P.
Bauer, Chas. A.
Blankenship, R. M.
Blessing, Jas. H.
Boenig, R. W.
Bole, W. A.
Bolton, R. P.
Bond, G. M.
Bonner, W. T.
Bourne, S. N.
Boyer, Francis H.
Brashear, J. A.
Brill, Geo. M.
Brooks, E. C.
Brown, Alex. T.
Bryan, Wm. H.
Bulkley, H. W.
Bullard, E. P.
Burbank, L. S.
Butcher, Jos. J.
Cassier, Louis
Castle, J. S. P.
Cheney, W. L.
Christie, Jas.
Cole, Winthrop
Coleman, Geo. F.
Colvin, F. H.
Comly, G. N.
Connell, J. A.
Cooper, H. R.
Coster, E. L.
Cowles, W. B.
Crain, J. J.
Cremer, J. M.
Darling, E. A.
Dashiell, B. J., Jr.
Davis, Daniel
Deane, C. P.
Detrick, J. S.
Doran, W. S.
Dowst, F. B.
Drewett, W. A.
Drysdale, W. A.
DuBosque, F. L.
Easby, Francis H.
Edwards, V. E.
Engel, L. G.
Emerson, H.
Fellows, E. R.
Felton, E. C.
Ferguson, G. R.
Firth, Wm. E.
Flagg, S. G.
Forbes, W. D.
Foster, Chas. E.
Freeman, Jno. R.
French, L. G.
Fritz, John
Frothingham, F. E.
Fryer, G. G.
Galloupe, F. E.
Geer, H. G.
Gifford, C. H.
Gleason, Wm.
Goodale, A. M.
Gordon, F. W.
Gould, W. V.
Granger, A. S.
Greenwood, P. F.
Griffin, C. L.
Grimm, P. H.
Grohmann, C. L.
Guest, J. J.
Gwilliam, G. T.
Hale, R. S.
Halsey, F. A.
Hand, F. L.
Harkness, Wm.
Hart, F. L.
Hartness, Jas.
Harris, F. A. W.
Hawkins, J. T.
Hayward, H. S.
Heggem, C. O.
Henderson, Alex.
Henning, G. C.
Herr, E. M.
Hillyer, Geo.
Higgins, C. P.
Higgins, M. P.
Holbrook, D. L.
Hoxie, W. D.
Huff, S. W.
Hunt, Chas. W.
Hutton, F. R., *Secretary.*
Jacobs, W. S.
Johnson, A. E.
Johnson, J. E., Jr.
Jones, C. R.
Jones, D. P.
Jones, F. R.
Kafer, J. C.
Keller, E. E.
Kellogg, J. W.
Kempsmith, Frank
Kendall, Jas. H.
Kent, Wm.
Kerr, C. V.
King, Frank B.
Kirkevaag, Peter.
Knickerbacker, John
Knight, Hervey S.
LaForge, F. H.
Lane, H. M.
Lawton, W. L.
Leitch, Meredith,
LeVan, W. B.
Lewis, D. J., Jr.
Lewis, Wilfred
Logan, Jno. D.
Low, F. R.
McBride, Jas.
McClelland, E. S.

McFarland, Walter A.
McFarland, W. M.
McMannis, Wm.
Mackintosh, Fred'k.
Manning, C. H.
Manning, H. G.
Mansfield, A. K.
Mathews, W. E.
Mattice, A. M.
Meier, E. D.
Melville, G. W., *President.*
Melvin, D. N.
Merriam, F. W.
Merriam, H. P.
Mesta, Geo.
Meyer, H. C., Jr.
Miller, E. E.
Miller, Spencer
Moore, C. A.
Moore, D. G.
Morgan, C. H.
Morgan, T. R.
Morton, G. L.
Mossberg, Frank
Muncaster, W. J.
Nagle, A. F.
Naylor, E. W.
Neff, E. H.
Newhall, J. B.
Nicoll, C. H.
Nichols, O. F.
Norris, H. M.
Noyes, Henry
Owens, R. B.
Parker, A. W.
Parks, E. H.
Paul, J. W.
Pierce, W. H.
Phillips, Franklin.
Phillips, G. H.
Pitman, S. M.
Porter, H. F. J.
Porter, Chas. T.
Pratt, C. R.
Raynal, Alfred H.
Reed, Sam'l G.
Reed, Wm. E.
Reist, H. G.
Reynolds, I. H.
Richmond, Geo.
Richards, F. H.
Richmond, K. C.
Richter, E.
Riddell, John
Rider, T. J.
Roberts, Wm.
Robinson, H. B.
Rockwood, Geo. I.
Rogers, W. S.
Rohrer, A. L.
Roney, Wm. R.
Ross, E. L.
Roux, Paul
Sabin, A. H.
Sargent, C. E.
Satherberg, C. H.
Schaum, O. W.
Schoenborn, W. E.
Schumann, Geo.
Scott, Jas. B.
See, Horace.
Serrell, J. A.
Seymour, Jos. W.
Simpson, Geo. R.
Smith, Geo. H.
Smith, Jesse M.
Smith, Oberlin.
Smith, S. Howard.
Snell, H. I.
Sparrow, E. P.
Spies, Albert.
Spilsbury, E. G.
Stanwood, J. B.
Stetson, G. R.
Stiles, N. C.
Stillman, F. H.
Suplee, H. H.
Swasey, Ambrose
Sweet, John E.
Swift, E. H.
Taber, Geo. H.
Tallman, F. G.
Thomas, E. G.
Tompkins, S. D.
Tompkins, Stonewall
Townsend, David
Tribe, James
Tyler, C. C.
Uehling, E. A.
Upton, Wm. B.
Varney, Wm. W.
Vaux, Wm. S., Jr.
Veeder, C. H.
Victorin, Anthony
Waldo, Leonard
Waldron, F. A.
Wallace, Wm.
Ware, J. A.
Warren, B. H.
Washburn, Wm. S.
Watson, Wm.
Webster, Hosea
Webster, Wm. R.
Wellman, Chas. H.
Whaley, W. B. Smith
Wheeler, F. Merriam
Wheeler, Seth
Whitehead, G. E.
Whitlock, R. H.
Wiley, Wm. H., *Treasurer.*
Williamson, Wm. C.
Willis, E. J.
Williston, A. L.
Willson, F. N.
Wood, A. J.
Wood, Walter
Wood, W. H.
Woolson, I. H.
Woolson, O. C.
Woolson, W. D.
Wood, F. W.
Wright, J. K.
Young, Wm. S.

The first order of business was the Report of Tellers of Election concerning the members seeking to join the Society at this convention. It was read by the secretary, as follows:

REPORT OF TELLERS OF ELECTION.

The undersigned were appointed a committee of the Council to act as tellers, under Article 14 of the Rules, to scrutinize and count the ballots cast for and against the candidates proposed for membership, in their several grades in the American Society of Mechanical Engineers, and seeking election before the XXXIXth meeting, Washington, D. C., 1899.

They have met upon the designated day, in the office of the Society, and have proceeded to the discharge of their duty. They would certify for formal insertion in the records of the Society to the election of the following persons, whose names appear on the appended list, in their several grades.

There were 477 votes cast on the blue ballot, of which 11 were thrown out because of informalities. The tellers have considered a ballot as informal which was not indorsed with an autographic signature, or where the endorsement was made by a facsimile or other stamp.

CHAS. H. LORING,
GUS. C. HENNING,
H. H. SUPLEE,
Tellers of Election.

AS MEMBERS.

Adams, Edward T.
Apps, Chas. H.
Aue, Joseph E.
Bell, Joseph M.
Bullard, Dudley B.
Bullard, Edward P.
Cathcart, Wm. L.
Corbett, Wm. H.
Crawford, David F.
Cummings, Orrie P.
Cunningham, Thos.
Daly, Michael J.
Davis, Daniel
Durand, Nelson C.
Emerson, Harrington
Ferguson, Hardy S.
Forstall, Alfred E.
Geer, Herbert G.
Guest, Jas. J.
Harris, Fred'k A. W.
Hillman, Cecil R.
Hoit, Lehman B.
Holbrook, Dio Lewis
Houghton, Chas. E.
John, Griffith
Kinne, Clarence E.
Klein, Otto H.
Kwang, Kwong Y
Loyd, John
MacLaren, Malcolm N.
Nickel, Franz F.
Nichols, Wm. Wallace
Rotter, Max
Sando, Will J.
Stearns, Chas. K.
Vaughan, Henry H.
Von Goeben, Carl K.
Wood, Frank J.

AS ASSOCIATES.

Emeny, Fred. J.	Raju, Sam'l D.	Yawger, Edwin
Murray, Chas. R.	Wood, Kenneth F.	

PROMOTION TO FULL MEMBERSHIP.

Ackerman, Wm. S.	Chamberlain, Paul M.	Smith, Chas. M. W.
Bissell, Geo. W.	Riker, Andrew L.	Williston, Arthur L.

PROMOTION TO ASSOCIATE MEMBERSHIP.

Hagar, Edward McKim

AS JUNIOR MEMBERS.

Barnay, John M.	Fernald, Benj. G.	Mason, Daniel A.
Caine, William P.	Grimes, Geo. L.	Mixter, Geo. W.
Castle, John S. P.	Hargrave, Russell Wm.	Moyer, Sam'l L.
Chandler, Irving A.	Hammers, Morgan J.	Seix, John, Jr.
Chapin, Warren W.	Heald, Geo. W.	Serrell, Wm. C.
Colt, Sam'l G.	Herbert, Fred'k D.	Sethman, Geo. H.
Cowperthwait, Allan	Hitchcock, Fred'k M.	Stevens, Robt. C.
Davey, Warren	Hollingsworth, Sam'l	Vanderbilt, Cornelius, Jr.
Dickerman, Wm. Carter	Hopton, Lemuel R.	Worden, Euclid P.
Dow, Carl S.	Knecht, Arthur E.	Young, John Paul
Eldred, Byron E.	McCallum, Alex. C.	
Fairbanks, Geo. A.	MacPherson, Jas. Day	

In view of the effort which was being made to increase the collections of the Society on account of dues, etc., with a view to making as large a purchase of bonds as possible, the Finance Committee had directed the preparation and publication of a semi-annual report. This report was as follows:

SEMI-ANNUAL REPORT OF THE FINANCE COMMITTEE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, NOV. 15, 1898—MAY 5, 1899.

For first half of the current year, 1898–9, the Finance Committee of the American Society of Mechanical Engineers would respectfully report to the Council and the Society the following statements of receipts and expenditures which have passed under their direction on behalf of the Society during the period beginning November 15, 1898, and ending May 5, 1889.

Secretary's Balance Sheet:

To Receipts for above mentioned period.....	$23,827.53	By Cash to Treasurer.....	$23,827.53

Itemized statement of Receipts and Expenditures of the Society for same period:

Receipts.			*Disbursements.*	
Initiation Fees		$810.00	Reprints and Publications	$3,773.07
Dues	Current. $18,859.73		Postage and Express	1,889.52
	Past 815.00	19,676.83	Salaries	3,855.00
	Advance. 2.10		Office Expenses	251.29
Pin Badges and Certificates		220.50	Engraving (illustrating papers)	1,654.30
Sales of Publications			Binding Transactions	1,694.10
	Volumes $711.75		Meetings	972.76
	Pamphlets. 58.95	775.45	Work of Committees	6.20
	Binding 4.75		Badges and Certificates	262.58
Engraving		71.80	Travelling	150.00
Rent	Hall $510.00		Insurance and Safe Deposit	60.00
	Rooms 1,057.08	1,567.08	Rent, Interest and Taxes	2,634.90
Life Membership		100.00	Printing Catalogues, Circulars, etc.	2,079.70
Office Expenses (Tel. Tolls)		3.80	Stationery Supplies	242.10
Interest on Investment (Bonds at $2.50)		595.00	House Supplies and Furniture	401.61
Misc. Accounts (all small sums)		7.07	Library (Book Purchase and Binding)	48.39
			Janitorial Supplies	76.43
			Fuel	22.50
			Lighting (gas and electric current)	257.43
			Laundry	156.50
			Repairs to House, Furn., etc.	608.79
			Collection Charges (Out of Town Checks)	1.03
			Interest on Investment, (Amount due on coupons attached to bonds bought at time of purchase)	15.01
			Total Disbursements	$21,111.21
Total Receipts		$23,827.53	Excess Receipts over Regular Disbursements	3,414.97
Cash on hand first of year		698.65		
Total		$24,526.18	Total	$24,526.18

Total Cash as Itemized above	$24,526.18	
" " Expended as Itemized above	21,111.21	
Excess Receipts over Regular Expenses	$3,414.97	
Expended for Purchase of $100 Bonds bearing interest @ 5%		$2,700.00
Cash in Treasurer's hands		714.97
Total	$3,414.97	$3,414.97

OUTSTANDING AND DUE THE SOCIETY, MAY 5, 1899.

Amount outstanding and due from 464 members, exclusive of new members elected May 4, 1899 (24 per cent. of membership)	$8,464.71
Amount outstanding and due from 16 non-members for publications, electros, rent, etc., etc	520.58*
Amount outstanding and due from 84 new members elected May 4, '99, for initiation fees and dues	2,655.00
Total amount due the Society this date	$11,640.29
264 bonds, owned by the Society, less one bond turned in for a life membership, which had its July, 1899, coupon cut off by former owner, equals 263 coupons at $2.50 each	$657.50
Total	$12,297.79

* $340 of this amount is due from one society, which rents our hall by the year, and this amount due represents what it owes us for the remaining months of the current year.

As an indication that the Society has felt the effect of the easier times throughout the country, it might be added that the collections at this period last year (1897–8) were $21,256.86, against $23,827.53 this year, a net increase of receipts this period this year over the same period last year of $2,570.67.

INVESTMENT OR BOND ACCOUNT.

Original issue of 5 per cent. interest-bearing bonds of Mechanical Engineers' Library Association which expire July 1, 1900	$32,000

PRESENT HOLDERS OF SAID BONDS.

Bonds held by the Council as Trustees for the Am. Soc. Mech. Engineers at end of year 1897–8	$23,600	
Bonds bought this year with surplus Society funds and so held	2,700	
Bonds received this year for Life Membership and so held	100	
Total Bonds so held by the Council as Trustees		$26,400
Bonds bought by the Mech. Engineers' Library Association and held by the Trustees of same	600	
Balance bonds still outstanding in hands of members	5,000	
		$32,000

It was estimated by the Finance Committee at the first of the year current that there would be $3,500 available this year for the purchase of these bonds, and as shown above we have expended to date $2,700 for such bonds, leaving $800 of that estimate still available for that purpose. If collections continue to be

as good as in the past for the balance of the current year, the Finance Committee hope to be able to expend more than the estimated sum for such bonds, and with this object in view it is urged that all members yet owing the Society should remit promptly, thus lending their aid toward this desirable end. It is the wish of the Council that the entire issue of these bonds should be acquired by the Society before the date of their expiration in July, 1900.

In its presentation, the Secretary urged upon all members the desirability of reducing the amount outstanding and due the Society, by prompt attention to statements of account which were to be sent to those whose accounts were still open in the early summer.

Under the heading of Motions and Resolutions, the Secretary read two letters inviting the Society to hold its spring meeting of 1900 in the city of Cincinnati, O. One of these was from the Cincinnati League, and the other from the mayor of the city. These invitations were seconded by Messrs. Lane and Baldwin, and a motion was put and passed that they should be referred to the Council, with the request that that body should give favorable consideration to the invitation from the citizens and mayor.

The professional committees of the Society were then called. The first was that upon the revision of the Code for Conducting Boiler Trials, which had been reported to the Society in 1885, and upon which the Committee had made provisional and tentative reports at the New York meeting of 1897, and again in 1898, and now presented with many of the suggestions incorporated, which had been offered in the discussions of previous meetings. The report, with printed discussion, was presented by Mr. Wm. Kent, who had been acting as Chairman of the Committee since the death of the late Charles E. Emery in 1898. Discussion was contributed by Professors Benjamin and Kerr, and by Messrs. Meier, Kinealy, Bryan, Hale, Suplee, and D. P. Jones. It was understood that the report and discussion would be referred back to the Committee for their consideration.

The Committee on Standard Methods of Reporting Steam Engine Trials reported progress through its chairman, Mr. F. H. Boyer. The work had taken somewhat of an international complexion from the presence of both British and American members, and replies and opinions had come to the committee in large volume from engineers in both countries. It was hoped that the

preliminary report might be presented for consideration by the Society at the fall meeting.

Pursuant to action taken at the New York meeting of 1898, a committee had been appointed to consider the advisability of enlisting the Junior membership in a larger share of the development of the Society. To this end a committee had been appointed, and a report from this committee was presented as follows:

Junior Meetings, Season of 1899.

The Junior movement to enlist a greater interest and participation in the work of the Society has now become fairly established. Two monthly meetings have been held, one in March and one in April, the first of which was well attended and the second of which taxed the capacity of the hall.

At the March meeting papers were read by Chief Engineer Gardiner C. Sims, of the *Vulcan*, repair ship of the United States navy, stationed with Admiral Sampson's fleet at Guantanamo Bay in the late war, and by Professor Aldrich, Passed Assistant Engineer on the same vessel. Mr. Sims gave a detailed account of the loss of the *Maria Teresa*, and Professor Aldrich of the equipment and work of the *Vulcan*. At the April meeting an illustrated paper was presented by Mr. Arthur L. Rice on the history of the compression and liquefaction of gases, with special reference to the liquefaction of air, supplemented by those experiments that have been so generally made public in this connection.

These papers have been put in suitable shape, and are filed in the library of the Society. Abstracts of them have been sent to each member, and a few full copies provided for circulation among those who wish them.

The committee that has this movement in charge consists of B. C. Ball, Arthur L. Rice, Percy Allan, F. E. Frothingham, and the Secretary of the Society, as advisory member. The committee outlined its creation and policy in a report made at the March meeting, which will be found in the Junior transactions. The following is a partial quotation from this report:

"We believe that this proposed movement is of much greater and more fundamental significance than at first sight appears. It is much more than a question of merely enlisting Junior participation and interest in the work of the Society. It has to do with

the future of this National Society of Mechanical Engineers, with the education and development of young engineers throughout the country, and with what should be of the first interest to every engineer: the raising of the standard of his profession, the helping to put it on the high level of professional work that it should occupy.

"It should be remembered that the body of Juniors will always supply the men who will represent the larger, real, Society in years to come, and the wider their experience and training, the more worthily can they do this. If the Junior members do not work together, do not strive to give each other all possible information and encouragement, and to take a more active part in the regular meetings of the Society, how can they expect the Society to continue to grow, to continue to have its transactions a necessary part of every engineering library, and to have those whose ability has put them among the Senior members honored in the profession? It goes without saying how such an effort will educate and develop us young engineers. We cannot meet regularly, read and discuss papers and exchange experiences, without learning something, and it will be a valuable opportunity to extend our acquaintance among the men we would like to know. And not only that, but we will enlist an outside interest never felt before among the younger men; can increase our membership, and so do better and better work; and as our members increase our influence increases, and the Junior section of the Society can make a name for itself if it will. It is fair to say that if we do make a name for ourselves, and it is a good one, that the Senior Society will benefit as much as we."

As this quotation indicates, the primary object of the movement was to get the Junior members of the Society together at short and regular intervals, on common ground and with common interests, to become mutually acquainted and to exchange views, so that interest in the larger work of the Society will be stimulated. Though Junior participation will still be one of the first objects sought, the unexpected interest manifested by Juniors and Seniors alike makes it possible for the committee to gratify its desire to broaden the scope of the movement, so that attendance and co-operation will be sought and will be most welcome from all. Monthly meetings will be held during the winter (December excepted) on the first Tuesday of each month, at the Society house in New York. They will be of such nature as circum-

stances seem to warrant, the idea being to have papers and discussions of interest and scientific worth presented. Special effort will be directed to making these meetings of more than local interest, and the committee is formulating a plan for reaching all distant members. To make the meetings of permanent value, transactions of each meeting will be kept on file in the Society library. These transactions will be examined by the publication committee with a view to having suitable papers presented by their authors at a regular meeting, in the regular way.

It should be understood that this movement is the result of effort on the part of the Council, and that in good behavior the Junior committees appointed shall direct its course.

So ready has been the response to the efforts of the committee, that the programme for the coming winter has already been determined upon. It was not easy for the committee to decide what of the proposed topics they could afford to leave for later meetings. The following subjects were finally selected as being of live and immediate interest: Piping, Cast Iron *vs.* Cast Steel, The Gas Engine, Bearing Metal Alloys, and Mechanical Stokers. The Gas Engine would probably take the time of at least two meetings. Definite announcements will be made later.

FRANCIS E. FROTHINGHAM,
Secretary Junior Committee.

At the close, a meeting of any Juniors who might be present at the Washington meeting was requested.

The President called again for any matters of business, but as no motions or resolutions were presented the meeting took up the consideration of the professional papers. The first was entitled "Standards for Direct-Connected Generating Sets," and was presented by J. B. Stanwood, of Cincinnati. In the discussion Messrs. Mansfield, Aldrich, Roberts and Forbes were heard, and at its conclusion the following resolution was put by the President, and as amended by the Council goes upon record in the following form:

"*Resolved*, That the Council be requested to communicate with the American Institute of Electrical Engineers to ascertain if that institute will agree to appoint a committee to co-operate with a similar committee, to be appointed by the American Society of Mechanical Engineers, to consider and report upon a standard series of capacities, speeds, and necessary dimensions for electrical generators for direct connection to steam engines. And, furthermore,

"*Resolved*, That if a favorable response be received, the President be requested, with the concurrence of the Council, to appoint a committee of five to co-operate with the committee to be appointed by the American Institute of Electrical Engineers, for the purpose herein set forth."

It may be added that the President subsequently appointed, under favorable action by the Council, this committee, to consist of Messrs. James B. Stanwood, W. D. Forbes, F. H. Ball, W. M. McFarland, and A. L. Rohrer.

The other papers of the morning were those by Messrs. Hale on "Boiler and Furnace Efficiency," and F. L. Emory on "Test of a Steam Separator," in whose discussion Messrs. Roney, Meier, Kent, and Boyer took part.

The President, before adjournment, read Article 31 of the Rules, in which he was directed to appoint a committee to nominate officers of the Society for the ensuing year. This committee is to report in advance of the annual meeting. It was made to consist of Messrs.

Charles Wallace Hunt, *Chairman*	New York City.
George H. Smith	Providence, R. I.
A. K. Mansfield	Salem, O.
Henry I. Snell	Philadelphia, Pa.
Walter M. McFarland	Pittsburg, Pa.

The President directed attention to the effort which had been made to distribute widely the membership of the committee among different sections, and on motion the meeting adjourned until the evening.

It had been the purpose of the Committee of Arrangements that on the afternoon of Wednesday the Society should be received by the President of the United States, Mr. William McKinley, in the Executive Mansion, after the hour at which this is open to the general public. By reason of the necessity imposed upon the President to secure rest and recuperation at the Virginia Hot Springs, the Society was compelled to forego the anticipated pleasure of a personal greeting. The White House, however, was open to the visitors, under the guidance of Col. T. A. Bingham, in charge of the public buildings, and an opportunity was given to visit the rooms which are usually not open to a visiting public. The conservatories were particularly enjoyed. Opportunity was taken on this afternoon for the taking of a group photograph upon the steps of the State, War, and Navy Building. After the completion of the visit to the White House, the party

took trolley cars for a visit to the Heights of Arlington and the Soldiers' Cemetery.

Third Session. Wednesday Evening, May 10th.

The professional papers of the Wednesday evening session were as follows: "Relation Between Initial Tension and Power Transmittal by a Belt," by F. L. Emory; "Experiments on Various Types of Fire Hydrants," by C. L. Newcomb; "Experiences with Deep-Well Pumping Rods," by G. W. Bissell; "Pipe Flanges and Their Bolts," by A. F. Nagle; "Manufacture of Car Wheels," by Geo. R. Henderson. In the discussion of the evening Messrs. Aldrich, F. R. Jones, McFarland, Lewis, Hutton, Kerr, Boyer, Halsey, Bolton, McBride, Kent, Suplee, J. E. Johnson, Jr., Melville, Rockwood, Hawkins, Meyer, Higgins, and Ashley took part.

Fourth Session. Thursday Morning, May 11th.

The papers of the morning were as follows: "The Equipment of Tall Office Buildings in New York City," by R. P. Bolton; "Rolling Mill Fly-Wheels," by John Fritz. The latter was moved forward from its previously assigned place, and displaced the regular papers of the morning. In the discussion on the tall buildings and the fly-wheel the participants were Messrs. Rockwood, Bryan, Ashley, Roberts, Kerr, Kent, Oberlin Smith, Henning, McBride, Lewis, Stanwood, F. R. Jones, Boyer, Brashear, Halsey, Comly, and Hawkins. Mr. Fritz was also persuaded to add his remarks to the discussion on Mr. Nagle's paper on pipe joints, at the close of this session. So much time had been consumed, however, that it was made necessary to move forward the discussion on power plants for large buildings or groups of buildings until Friday morning.

On the afternoon of Thursday the party, by invitation, visited the big gun factory of the Navy Yard and the buildings attaching to the Ordnance Museum, but special interest centered around the experimental model tank which had recently been completed. The visitors were permitted to inspect in detail the bridge and its motor machinery, the dynamometer appliances and their recording attachments, and the provisions for the safety arrest of the moving bridge at high speed. They were also escorted to the

shop where the models were made, and were shown the special machinery used in reduplicating the lines. After the visit was completed, those who desired were conducted by the cars of the traction company, without charge, to the other end of the city for a visit to the power-house of the company.

In the evening the visiting members and their ladies were the guests of Mrs. George Westinghouse, in the Blaine mansion on Dupont Circle. A great number of representative residents of Washington, both in its political and official circles, had been invited to meet the Society, and to furnish accommodation an extra structure had been erected to form a ball-room at the rear of the house. A second orchestra was established in this ball-room for the enjoyment of those who were out of hearing of the principal orchestra upon the landing of the stairs. It was a pleasure to see and greet persons whose names had become familiar by reason of their achievements during the recent war. The orchids and the decorations were upon a scale of beauty that will long be remembered.

Fifth Session. Friday Morning, 12th.

This session was called at an earlier hour than usual, by reason of the disturbance in the programme which had been caused by the discussion of fly-wheels on the previous day. The papers of the morning covered the following: Storm Bull, "Central Heating Plant of the University of Wisconsin"; E. A. Darling, "The Power Plant of a University"; G. I. Alden, "The Plunger Elevator"; C. R. Pratt, "Elevators"; C. H. Quereau, "The Allen Valve for Locomotives"; F. W. Gordon, "A New System of Valve for Steam Engines, Air Engines and Compressors." The participants in debate were Messrs. Bolton, Bryan, Kent, F. R. Jones, Robinson, Aldrich, Boyer, Rogers, Oberlin Smith, Wm. J. Baldwin, P. H. Grimm, Le Van, and Woolson.

At the close of the papers the concluding business of the convention came in order, and the consideration of motions and resolutions. The Secretary read from the Council minutes the following resolutions:

Resolved, That the Association, as a body and through its individual members, urge upon Congress the necessity of relieving the present overcrowded condition of the Patent Office, and providing sufficient room, force, and facilities for the prompt and proper execution of its work.

Resolved, That we further urge that the records of the office, which so largely

constitute the legal evidence of title of so many of the large manufacturing industries of this country, should be more safely stored, and that ample appropriations be made for providing incombustible receptacles for the records.

Resolved, That we especially urge that the library of the Patent Office, upon which the efficiency and accuracy of the work of the bureau depend, shall have such ample appropriation for its extension in its special field, and for keeping it fully abreast of the progress in the mechanical and manufacturing arts of the day.

Resolved, That this Association urge the necessity of giving to the Patent Office the use of the entire building in which it is now located, and that the moneys paid into the Patent Office by inventors be applied so far as necessary to the uses of the office.

Mr. Hunt rose to second action upon these resolutions, and spoke as follows:

Mr. C. W. Hunt.—In rising to present motions urging the action of the Society upon these resolutions, I would call the attention of the Society to the fact that this Government has two sources of income: one is that which comes from taxation, direct and indirect, and the other is the income from its Patent Office. The first resolution I will offer is:

Resolved, That the American Society of Mechanical Engineers hereby approve and confirm the action of the Council of the Society in urging Congress to pass such laws and make such provisions as will enable the Patent Office to dispatch the business of the bureau in an effective and accurate manner.

The second resolution is:

Resolved, That we recommend that our individual members be requested to bring the wisdom and justice of the action proposed to the personal attention of our public officers as the way opens to them.

Mr. Ashworth.—I move the adoption of the resolutions.

The motion was carried.

The Secretary then called the attention of the meeting to the precedent which had seemed to work well at previous meetings, whereby the Council should appoint a committee of members to draft the resolutions of thanks, whereby the deliberative body should put on record its appreciation of the courtesies extended to them during their visit in the convention city. He held in hand the report of such a committee in the form of a series of resolutions which were read, seconded, and enthusiastically adopted. They carried with them the instructions to transmit them to those interested. The resolutions were as follows:

The members of the American Society of Mechanical Engineers, at the close of this most successful Washington meeting, desire to put on record their sincere

appreciation of the courtesies received at the hands of the Board of Trustees of the Corcoran Art Gallery. The opening reception in the spacious halls of that structure was an earnest of the brilliancy and success which was to attach to a memorable meeting of the Society.

In this connection the Society wants to say to Col. T. A. Bingham, United States Engineers, that his address of welcome on that evening formed a fitting prelude to the generous courtesy with which the City of Washington has received the visiting engineers.

Resolved, That while it is sometimes desirable that the left hand of a Society should not know what its right hand is doing, yet the Society has learned with pride and pleasure that to its honored President, Rear-Admiral George W. Melville, Engineer-in-Chief, U. S. N., the Society owes a debt of pleasure for his share in making that evening brilliant and successful.

It is not always easy to return thanks to yourself, but the President will understand what the Society would like to say, now that it has a chance to do so.

Resolved, That the thanks of the Society are due to the Curator of the Gallery, Dr. F. S. Barbarin, for his admirable arrangements for the comfort of the visitors, and for the most delightful way in which his responsible duties have been discharged.

Resolved, That the sincere thanks of the American Society of Mechanical Engineers are due and extended to President William McKinley, and to his efficient aides in charge of the Executive Mansion, for the invitation to visit the White House under such favorable circumstances. The Society can only express its keen regret and sympathy that the occasion of the President's absence from the city for the sake of his health should have had to be the reason which has prevented the members and their ladies from greeting the President in person on the occasion of this reception. They express the hope that the rest will bring a speedy return to health and strength for the responsible duties of the presidential office.

The formal mould of preamble and resolution, whereby a deliberative body tries to put on record the feelings which animate the individual members of that body, is an unsatisfactory vehicle when a depth of appreciation and a warmth of recognition are to be conveyed such as animate the Society of Mechanical Engineers in its memory of the reception Thursday evening, May 11th, tendered to the Society by Mr. and Mrs. George Westinghouse, of Washington.

The engineers are themselves familiar with what it means to arrange undertakings which involve immense detail, and for this reason are the more appreciative of the result when they see successful planning by others, and when that planning is for them. The beauty of the Blaine mansion and its treasures would have been a pleasure in themselves, but the decorations and the art evinced in the special construction planned by Mr. and Mrs. Westinghouse, together with the most admirable arrangements for the handling of an unusual number of visitors, will form a pleasant memory of our Washington meeting which it will take long to dim.

The Society appreciates, also, the opportunity of meeting under such favorable circumstances so great a proportion of the delightful citizenship of the Federal Capital, and the opportunity of taking by the hand many whose names belong to the history of our country in the closing years of the century.

The Society request, therefore, that in the following action their host and hostess will read between the lines a warmth of real feeling which is inadequately conveyed in the following resolutions:

Resolved, That the Society of Mechanical Engineers instruct its Council to convey to Mr. and Mrs. Westinghouse, by a personal expression, the thanks of the Society for their reception during the Washington meeting.

Resolved, That the Secretary be directed to have these resolutions engrossed in proper form for transmittal to Mr. and Mrs. Westinghouse, together with a proper letter accompanying them.

Resolved, That the American Society of Mechanical Engineers has appreciated most heartily the opportunity which was extended to it on Thursday afternoon to visit the Navy Yard and its varied interests. They would ask that their thanks may be coupled with a special expression of recognition to Commandant A. H. McCormick, of the Navy Yard, and to Admiral Charles O'Neil, of the Ordnance Department. They would also express to Naval Constructor David W. Taylor a most appreciative word for the exhibition of the technical details of the experimental model tank and its accessories, and for his own courtesy and assiduity in his attention to the visiting members.

The American Society of Mechanical Engineers desires to express its recognition of the special courtesy extended to the members with respect to opportunities for a visit, under such favorable circumstances, to the Washington Monument.

They recognize the kind hand of their friend, Col. T. A. Bingham, in charge of public buildings and grounds, in the admirable arrangements which have been made for the pleasure of the visitors during their stay.

Resolved, That the Society of Mechanical Engineers, upon the adjournment of this Washington meeting, desire to express to Chief Engineer George W. Baird, U. S. N., their sincere thanks for the arrangements which he has made for visits by the members to the State, War, and Navy Building, for the providing of guides and other facilities. It is impossible in this resolution to include in detail all the courtesies which have been enjoyed by the members individually at the hands of individual representatives of these departments. The best that the Society can do is to ask that any failure of theirs to give special and personal recognition means that they have received so much of kind service that it has become impracticable to give anything like adequate expression of thanks.

The American Society of Mechanical Engineers desires to thank Mr. Franklin W. Smith, Curator of the Halls of the Ancients, for the special courtesies and facilities extended to our members and their ladies on their visits to these interesting exhibits.

The thanks of the American Society of Mechanical Engineers are extended to the authorities of the Library of Congress, and particularly to Mr. Bernard R. Green, Superintendent, for courtesies enjoyed during their stay in Washington.

There is, perhaps, no department of the governmental service in which the manufacturers and mechanical engineers of America feel a keener interest, or whose importance they recognize more fully, than that of the U. S. Patent Office. In this view they want to extend to Charles H. Duell, Commissioner, and the Assistant Commissioner, Arthur P. Greeley, and their assistants, their thanks for the courtesies in connection with the visits which have been paid to the Patent Office by members of the Society. It is the hope of the body that as the result of better knowledge of the limitations under which the Patent Office is struggling to do its great work, some action may be taken which will secure the U. S. Patent Office some recognition and increased facilities for the work upon which it is engaged.

Resolved, That the American Society of Mechanical Engineers desire to express to Prof. C. F. Marvin, and all other officials of the U. S. Weather Bureau, their

thanks for the complete arrangements made by that bureau for the visits of those interested in its manifold duties.

The American Society of Mechanical Engineers, on the adjournment of its Washington meeting, desires to put on record its hearty thanks to Prof. Henry S. Pritchett, Superintendent of the U. S. Coast and Geodetic Survey Office, and to those connected with it, for their courtesy and arrangements in their behalf.

Resolved, That the thanks of the Society are due and extended to Prof. William Harkness and to his competent aides in charge of the U. S. Naval Observatory for complete arrangements for the pleasure of those who had the opportunity to visit this distinguished centre from which radiate lines of interest all over the country, having a most important technical bearing.

Resolved, That the American Society of Mechanical Engineers desires to thank Mr. George M. Bowers, Commissioner of Fish and Fisheries, for the special opportunity to visit the aquarium and other display, showing the important work upon which they are engaged. Their thanks are due in particular to Past Assistant Engineer F. S. Bowers, U. S. N., who was so courteous to those visiting this installation.

The American Society of Mechanical Engineers take occasion to express to Mr. Walter Ashfield McFarland, and the other officials connected with the Washington Water Works their thanks for the courtesies of their invitation to visit its interesting plant, and for the attentions which have been paid to those of the members who have been able to accept the invitation.

Resolved, That the American Society of Mechanical Engineers extends to F. W. Palmer, Public Printer, and to his associates in the Government Printing Office, its sincere thanks for the cordial invitation to visit the power plant of that office, upon the occasion of its recent meeting.

This invitation is appreciated none the less by reason of the fact that the pressure of other engagements has prevented any considerable number from availing of it.

Resolved, That the thanks of the American Society of Mechanical Engineers are extended to Dr. E. M. Gallaudet for his courteous invitation to the visiting members to witness the exercises of the College for the Deaf. The members regret that the pressure of other opportunities upon their limited time during the Washington visit has made it impossible to avail to any extent of many of the opportunities which have been put within their reach.

To the Presidents and officials of the street railroad companies of Washington the Society of Mechanical Engineers wish to express their sincere thanks for the courtesy of a free transportation and special arrangements for their comfort, and for the privilege of a most interesting visit to their respective power stations during the time of their stay.

The American Society of Mechanical Engineers desires to express to the Board of Governors of the Army and Navy Club, and to W. H. H. Southerland, Secretary, its sincere thanks for the privileges of the club extended to members during the period of the session of the Society in Washington.

It has been a feature of this meeting that the opportunities which Washington has placed within the reach of members of the Society have been so numerous that it has not been practicable to avail of many considerable attractions.

To the Board of Governors of the Cosmos Club the American Society of Mechanical Engineers desires to extend a most appreciative vote of thanks for their action in extending to the visitors in Washington the courtesies of the club house during the days of their stay.

The many attractions of the city have filled so full the time which the visitors

could spare from their professional sessions that it has been possible for a limited number only to avail of the privileges which had been so courteously extended.

To Major Richard Sylvester, Chief of Police of the Washington Police Force, the American Society of Mechanical Engineers extends its hearty appreciation for the admirable police arrangements which have added so much to the smooth working of the Washington meeting of the visiting engineers.

Among the special and individual courtesies which Washington alone could render to the visiting body is the opportunity for the visit to the art treasures which have been gathered together in this central city. The members and their ladies who have enjoyed the privilege of visiting the gallery of Mr. Thos. E. Waggaman desire to express to him their sincere and appreciative thanks for the pleasure he gave them by his kind interest.

The American Society of Mechanical Engineers desires to thank the Local and Executive Committees for their thoughtful methods and able, untiring efforts, which could not but compel success and have resulted in making this convention, the largest ever held, a most memorable occasion in our history.

The perfection of their arrangements, the judicious distribution of the time between business meetings, excursions to points replete with interest to engineers and patriots alike, and social gatherings warm with true Southern hospitality, made our stay one round of enjoyment, in which it were difficult to tell whether profit or pleasure predominated.

The American Society of Mechanical Engineers desires to thank the Committee of Ladies for the bright welcome extended to our wives and daughters, and the charming manner in which their duties as guides were performed, adding a living interest to the archæological, historical, scientific, and artistic collections for which their beautiful city is famed.

In acknowledging the resolution, on behalf of the Local Committee its chairman spoke as follows:

Mr. Frank B. King.—In acknowledging the Society's vote of thanks, as chairman of the Executive Committee there is one thought which, I believe, all the resident members desire to leave with you.

Before the name of Washington was firmly fastened upon this capital it was widely known throughout the States as "The Federal City." There was a significance to our grandfathers in that name which we have in a measure lost, and we must regret that it should be so. Washington has suffered from nothing so much as the unwillingness of the population at large to recognize in it the Capital of a great country. Our City Council is the Congress of the United States, but each incoming Congressman has to undergo years of training before he understands that he has obligations in this respect. Indeed, he is more apt to regard the District of Columbia as an experimental station, where all things of doubtful utility may be tested before they are applied to his precious constituency.

I doubt whether a majority of our visitors, though they fairly represent the thinking class of the population, have ever given this matter attention, and I would ask you, in leaving us, to spread this thought; and to move many of your associates in the circles in which you are, as a rule, so influentially connected, to come frequently to Washington, to glory in its development, and in a truly national spirit direct its growth.

Washington and the Society can help each other much. There are the strongest reasons why our conventions should be held here at stated intervals, and be looked forward to as regularly as those in New York. They might be three years apart; they might be four or five years, but they should not be twelve years. The idea of periodicity must not be lost, and no Washington meeting should be set aside, except for a World's Fair, or some supreme professional attraction. We have shown you that no other city can bring out such an overwhelming gathering of our members, and that must always be one of the main objects of our conventions.

I would entwine, if possible, yet more closely the success of the Society and the progress of the nation, and as a nation's progress finds its idealization in the complexion of its capital, Washington should be the object of your frequent pilgrimage and sincere solicitude.

I hope, then, that our convention here does but inaugurate an endless series of more and more brilliant occasions, and that long after we are gathered to our fathers the Society shall still be holding its periodical conventions in this then resplendent city, "the eternal Capital of an eternal Nation."

The President thereupon announced that a motion to adjourn would be in order, which was made, put, and carried.

The closing official event of the convention programme was the excursion of the Society by steamer to Mount Vernon and the tomb of Washington. This was set for the afternoon of Friday. Pursuant to the policy on which the Society aims to conduct its meetings, the tickets for this boat trip were purchased by the members, an addition of a few cents being made to the price of each ticket by the Local Committee, in order that a wreath might be presented by the Society on the occasion of its visit to the tomb, and laid with fitting ceremony as a mark

of respect from the Society. Appropriate remarks were made by ex-President C. W. Hunt at the grave of Washington, and the wreath was attached to the grating. Members then visited the historic mansion and its collections, and returned by boat to the city.

An interesting feature of the visit of the Society to Mount Vernon took the form of an aftermath on the morning of Saturday, May 13th. Mr. A. H. Raynal, Secretary of the Committee of Arrangements, planted a fine white oak sapling near the tomb, in the name of the Society, as a memorial of their visit. The ceremony was performed in an appropriate and dignified manner, in the presence of the custodian of the grounds, Mr. Harrison Howell Dodge, and of Mrs. Mitchell, who acted as sponsor. Mrs. Mitchell is the granddaughter of the Dean of the Regents. The tree will be one of a row of memorial trees on the driveway that leads from the tomb to the mansion. It is numbered six, on the left-hand side of the driveway, about 100 feet from the tomb. The Board of Regents of the Mount Vernon Association were in session at the mansion, and expressed the thanks of the Association to the Society for the handsome floral tribute, and the tree thus passed into the responsible possession of the authorities.

In the evening, as well as on the evening of Wednesday, the Washington Monument was made a special feature for the interest of the members. Col. T. A. Bingham and his representatives were present to explain to visitors the features of interest visible from the top of the monument Later, also, under the competent guidance of Mr. Bernard R. Green, Superintendent of the Congressional Library, the members were privileged to visit the ligh ing and ventilating machinery of that beautiful building, as well as the mechanical arrangements for conveying books from the library to the committee rooms at the cap tol, and the distributing system in the library itself. The visit to the reading room, and the explana ion of i s adminis ration, were made to the members after the general public had been dismissed from it by the close of the reading hours, so that the visi ors had full opportunity to see all that they desired without interference.

Besides what has been mentioned in the regular schedule, there was much of individual and semi-personal attention ex tended to the Society, of which the resolutions of thanks make

mention, but which were not a feature of the official programme. Special mention might be made of the courtesies of the important clubs, the Waggaman Gallery, the Hall of the Ancients, etc.

The suggestion made by Mr. King met with a certain emphasis of general approval, that the policy of holding meetings periodically in the national capital would be one which would not only be very agreeable to the members but would also be of advantage to the profession and to the city of Washington, as well as emphasizing the distinctly national character of the organization. The question will be brought up and considered in the Council.

DCCCVIII.*

RELATION BETWEEN THE INITIAL TENSION AND POWER TRANSMITTED BY A BELT.

BY F. L. EMORY, MORGANTOWN, W. VA.

(Member of the Society.)

By an examination of the diagram (Fig. 146) it would appear that the tension on the belt decreases gradually along the arc EBD from the value T_1 at E to T_2 at D. Let it be T at B and $T + dT$ at C. The portion CB of the belt, when the latter is at the point of slipping, is then under the action of the forces

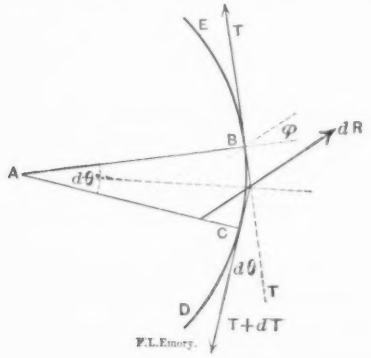

Fig. 146.

T_1, $T + dT$, and of the reaction dR; therefore the reaction must pass through the intersection of the components and makes an angle with the radius equal to the coefficient of friction. Resolving these forces along T and at right angles to it, and we have $T + dR \sin \varphi = T + dT$, $dR \cos \varphi = (T + dT) \sin d\theta = (T + dT)\, d\theta$, and, if along and at right angles to $T + dT$, then $dR \sin \varphi = dT$, $dR \cos \varphi = T d\theta$, dividing, and $\tan \varphi = \dfrac{dT}{T d\theta}$, since $\tan \varphi = \mu$, then $\dfrac{dT}{T} = \mu d\theta$, and, integrating, $\log \dfrac{T_1}{T_2} = \mu\theta$, or $\dfrac{T_1}{T_2} = e^{\mu\theta}$, or $T_1 = T_2 e^{\mu\theta}$, and $\therefore\ T_1 - T_2 = T_2 (e^{\mu\theta} - 1)$.

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

To find the initial tension on a belt when still, in order to transmit or develop a certain work when running.—It is evident that the total pull tending to shorten the distance between the shafts, or the total stress along the line of centres of

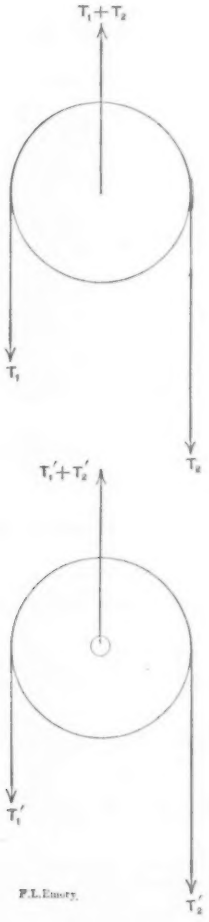

FIG. 147.

shafts, is the same when running as when still, and the only change is in the difference in tension between the two sides. If this be true, and T_1 and T_2 be the tensions at a given speed to develop a certain horse-power, and if, further, T_1^1 and T_2^1 be the tensions when at rest, then evidently $T_1^1 + T_2^1 = T_1 + T_2$

(Fig. 147). Now, neglecting the friction on the bearings in the discussion, the tension on the two sides of the belt when still is the same, or $T_1^1 = T_2^1 \therefore 2T_1^1 = T_1 + T_2$, or $T_1^1 = \frac{T_1 + T_2}{2}$. Now, since by previous formula $T_1 = T_2 e^{\mu\theta}$ by adding T_2 to both sides $T_1 + T_2 = T_2(e^{\mu\theta} + 1)$, and since $T_2 = \frac{T_1}{e^{\mu\theta}}$, $T_1 + T_2 =$

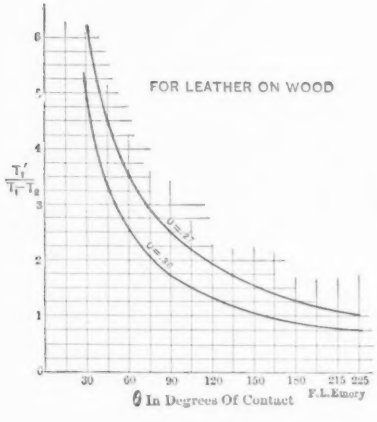

FIG. 148.

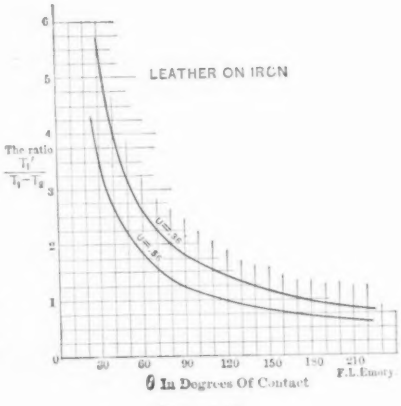

FIG. 149.

$\frac{T_1}{e^{\mu\theta}}(e^{\mu\theta} + 1)$; again from previous formula, $T_1 = \frac{(T_1 - T_2)e^{\mu\theta}}{(e^{\mu\theta} - 1)} \therefore$ $T_1 + T_2 = T_1 - T_2\left(\frac{e^{\mu\theta} + 1}{e^{\mu\theta} - 1}\right) \therefore T_1^1 = \frac{T_1 - T_2}{2}\left(\frac{e^{\mu\theta} + 1}{e^{\mu\theta} - 1}\right)$.

To use this formula, then,

$$T_1^1 = \frac{T_1 - T_2}{2}\left(\frac{e^{\mu\theta} + 1}{e^{\mu\theta} - 1}\right), \text{ or } \frac{T_1^1}{T_1 - T_2} = \frac{(e^{\mu\theta} + 1)}{2(e^{\mu\theta} - 1)}.$$

Since $T_1 - T_2$ is the pull which does work of rotation, then the initial tension is given in terms of the pull necessary for the work, and the coefficient of friction and arc of contact. When the arc of contact is greater than π on one wheel, it is necessarily less on the other, for straight belts.

Graphical Illustration of the Use of this Formula.—If we now substitute in the above various values for μ and θ in π measure, we obtain various values for the ratio $\frac{T_1^1}{T_1 - T_2}$, thus

FIG. 148.—FOR LEATHER ON WOOD, WET. (FOR $\mu = .27$.)

Radians $\theta =$	.5236	.785	1.047	1 57	2.355	3.1416	3.9245
$\frac{T_1'}{T_1 - T_2}$	6.1	4.8	3.5	2.48	1.6	1.25	1.03
$\mu\theta$ Radians.	.1416	.212	.282	.424	.636	.848	1.057
$\theta°$ Degrees.	30°	45°	60°	90°	135°	180°	225°

FIG. 148.—FOR LEATHER ON WOOD, DRY. (FOR $\mu = .38$.)

θ in Radians.	.5236	.785	1.047	1.57	2.355	3.1416	3.9245
$\frac{T_1'}{T_1 - T_2}$	5.	3.39	2.55	1.7	1.2	.93	.709
$\mu\theta$ Radians.	.199	.298	.3978	.596	.894	1.192	1.491
° Degrees.	30°	45°	60°	90°	135°	180°	225°

FIG. 149.—LEATHER ON IRON, WET. (FOR $\mu = .36$.)

θ in Radians.	.5236	.785	1.047	1.57	2.355	3.1416	3.9245
$\frac{T_1'}{T_1 - T_2}$	5.6	3.56	2.67	1.82	1.25	.97	.8
$\mu\theta$ in Radians.	.1785	.282	.337	.5652	.848	1.13	1.413
θ in Degrees.	30°	45°	60°	90°	135°	180°	225°

FIG. 149.—LEATHER ON IRON, DRY. (FOR $\mu = .56$.)

θ in Radians.	.5236	.785	1.047	1.57	2.355	3.1416	3.9245
$\frac{T_1'}{T_1 - T_2}$	3.57	2.31	1.85	1.213	.91	.7	.624
$\mu\theta$ in Radians.	.283	.44	.546	.88	1.32	1.76	2.198
θ in Degrees.	30°	45°	60°	90°	135°	180°	225°

To use the curve:

Problem—Find the initial tension on a belt to transmit 50 horse-power if the belt speed is 4,000 feet per minute. Since H.P. $= \frac{FD}{33000}$, then $50 = \frac{F \times 4000}{33000}$, $F = 412.5$; that is, the force performing the work is F, and this corresponds to $T_1 - T_2$, or $T_1 - T_2 = 412.5$ pounds. Now, if from the drawing showing distances between pulleys and sizes of pulleys it is found by the protractor or otherwise that the arc of contact is, say, 155 degrees, then for leather on wood, using the coefficient of friction of .38, we find the ordinate corresponding to the point on the curve having an abcissa of 155 is 1.1, about; then $\frac{T_1'}{T_1 - T_2} = 1.1$, and, substituting the value of $T_1 - T_2$ from above, and $T_1' = (T_1 - T_2)\,1.1 = (412.5)(1.1) = 453$ pounds; therefore the initial tension should be 453 pounds.

The values of μ here taken are the limiting values for dry or wet condition, and evidently the curves with values between the ones figured would be parallel to, or, in other words, lie between these drawn, and for values different from those given one may by interpolation select values for $\frac{T_1'}{T_1 - T_2}$ to suit his judgment.

DISCUSSION.

Mr. Wilfred Lewis.—When the arc of contact, θ, and the coefficient of friction, μ, are given, it has been shown by Rankine & Weisbach, and again in the paper now under discussion, that the ratio of the tensions $\frac{T_1}{T_2} = e^{\mu\theta}$. But the ratio of the tensions as thus expressed is really a limit not to be exceeded for the assumed values of θ and μ, and as a matter of fact $\frac{T_1}{T_2}$ may have any value less than $e^{\mu\theta}$ for which a new value of μ might be calculated. It may, however, be questioned whether the whole arc of contact, θ, is in every case called into action as a driving surface. In some part of the arc of contact the belt tension necessarily changes from T_2 to T_1 or from T_1 to T_2, and in making this change the belt must lengthen or contract, and consequently slip upon the pulley surface. When the percentage of slip is greater than the percentage of elongation in the belt due to the change

of tension, the whole arc of contact is necessarily engaged in the work of driving, but when these percentages are equal, as they should be to avoid unnecessary loss of power, the arc of contact may or may not be wholly engaged in driving, and a value of μ determined from θ, and the ratio $\frac{T_1}{T_2}$ might be misleading.

The coefficient of friction, μ, is therefore a very elusive quantity, and experiments have shown it to depend more upon the actual velocity of sliding than upon the percentage of slip, so that a belt may be quite efficient at full speed, and yet utterly fail under the same load at slow speed. Examples of this are not hard to find in practice, and I may say this was one of the unexplained facts that led William Sellers & Co. to make their experiments on the transmission of power by belting. The correctness of the well-known formula $\frac{T_1}{T_2} = e^{\mu\theta}$ may, I believe, be questioned not so much on account of increasing and excessive amounts of slip, as suggested at the previous meeting by Professor Aldrich, as on account of the variable rate of sliding due to the normal extension or contraction only of the belt upon the pulley surface in transmitting power. Excessive slip tends to make the rate of slip more nearly uniform over the whole arc of contact, and therefore it would seem to sustain the assumption of a uniform coefficient of friction at any given instant. On the other hand, if the belt does not slip over the whole arc of contact, the velocity of sliding must vary from nothing, at some point in the arc, to a maximum indicated by the percentage of elongation times the belt speed. This variable velocity of sliding cannot fail to produce corresponding changes in the coefficient of friction which vitiate to some extent the correctness of the formula in question.

But, whatever the coefficient of friction may be, it has been shown by the Sellers and other experiments that under normal working conditions the relation $T_1 = 3T_2$ may be safely realized in practice. The belt is then neither too tight to waste power in excessive journal friction, nor too loose to drive properly without danger of excessive slip, and a reasonable margin of safety is assured for atmospheric changes affecting the tensions and the surface condition. For this relation of the tensions we have $T_1 + T_2 = \frac{4}{3}T_1'$ and, neglecting the well-known increase in the sum of the tensions when driving, the initial tensions $T_1' = T_2'$ would appear to be $\frac{2}{3}T_1$, or in terms of the working tension

$T_1 - T_2 = \frac{2}{3} T_1$ we have $T_1^1 = T_1 - T_2$ (1). This very simple rule, to make the initial tension equal to the working load, may be applied to belts at moderate speeds with the comfortable assurance that some margin is provided for the stretching and loosening that is sure to follow, nor is it necessary to know exactly the angle, θ, if it approximates 180 degrees, as is generally the case.

But there is another important consideration which applies particularly to high speeds, and remains unnoticed in the paper under discussion.

The effect of centrifugal force upon the active tension of a belt is well known, and may be expressed by the equation $T = .013\ V^2$, in which T = centrifugal tension in pounds per square inch, and V = velocity of belt in feet per second, and if we let T_1^1 T_1 and T_2 also express tensions per square inch of belt section we should, to allow for centrifugal force, make $T_1^1 = (T_1 - T_2) + T$ (2). In the example taken by Mr. Emory of a belt running 4,000 feet a minute, we have $T = 57.8$ pounds, and if his belt section is assumed to be 2 square inches, clearly 115.6 pounds should be added to his result.

The important influence of centrifugal force upon the power transmitted by a belt was elucidated in one of the earliest papers read before this Society,* to which reference has been frequently made, but the initial tension to which belts should be laced in practice has not, I believe, been very clearly formulated.

It should always be somewhat more than just enough, for which equation (2) is a free expression to avoid the necessity for frequent tightening, and for those who are willing to accept this equation as a satisfactory approximation another relation may yet be drawn.

In another form, equation (2) may be written $T_1^1 = \frac{2}{3} T_1 + T$ (3), and in the proper use of belting T_1 should have a limiting value as a safeguard against undue stretching and rapid deterioration. At the same time the initial tension T_1^1 should not exceed T_1, and consequently the centrifugal tension T should not exceed $\frac{1}{3} T_1$.

In this way the safe working tension can be used to limit the speed beyond which a belt should not be driven. For example, if $T_1 = 400$ for cemented belts without lacings, we have V = about 100 feet per second or 6,000 feet per minute, a limit which is also

* "Horse-power of Leather Belts," by A. F. Nagle.

indicated for the speed of maximum power. So also for laced belts at 275 pounds per square inch, the maximum velocity is found to be about 5,000 feet per minute, the same as indicated for maximum power. Some reduction might be made in the initial tension, as expressed by equation (3), if time were taken to consider the probable increase in the sum of the tensions arising from the length and position of the belt, as well as from the peculiar elastic property of leather, but this reduction would apply chiefly to long horizontal belts which may require extra tension to prevent undue sag, and which, by reason of their length, may stretch without serious inconvenience.

As a rule for initial tension, equation (3) appears, therefore, to be on the safe side, and it is here offered as a practical solution of a problem on which much labor may be spent to little purpose. It provides simply a moderate excess of initial tension, which for obvious reasons is an absolute necessity in successful practice.

*Mr. Emory**.—I am indeed glad that Mr. Lewis has taken occasion to discuss the topic at some length. It was presented with the hope of promoting discussion that might reveal a purely theoretical, analytical treatment which conforms to experiment, and which would have little of the empirical about it.

As Mr. Lewis says, "the initial tension to which belts should be laced in practice has not, I believe, been very clearly formulated." It was this same belief on my part that caused me in part to bring up the subject. The theory as presented is not new, although it is given in the latest works on theoretical mechanics. It takes no account of the centrifugal force due to velocity, neither do any of the analytical demonstrations in our text books, nor does it take account of the change in μ due to slip of stretch. These facts put the teaching of theoretical mechanics at times into undesirable positions, to avoid which a direct, analytical, truthful treatment consistent with experiment is necessary—one which a student may readily see, and which demonstrates its truthfulness by taking into consideration all the conditions.

I do not wish to be considered as agreeing with the above theory as a finished product. Any one comparing the figured results with the best practice can see that either one or the other should be revised. Evidently the theory should give way to a more complete one. In these times a theory is not complete unless

* Author's closure, under the Rules.

it contains all the actual conditions of the practical case, and the above example proves this very satisfactorily.

A number of most excellent papers on the practical data of belts, leading to empirical formulæ, have been presented to the Society, but I am not aware of any analytical demonstration leading to a complete theory in which all these conditions are incorporated. Expressions may be derived including the change in the coefficient of friction of motion due to stretch, and of the effect of centrifugal force, and so on, but they lead to differential equations, which thus far I have not been able to integrate, owing to our ignorance of the manner of change of μ. It is with the hope of finding such a solution, together with the above mentioned reason, that this subject is introduced.

DCCCIX.*

THE ALLEN VALVE FOR LOCOMOTIVES.

BY C. H. QUEREAU, DENVER, COL.

(Member of the Society.)

THERE are decided differences of opinion among railroad mechanical men as to the value of the Allen, or double-ported, valve, a diagram of which is given in Fig. 150. Those who favor its use claim that it secures a greater mean effective pressure than the plain valve by giving a more prompt and full admission, thus increasing the power of locomotives on which it is used.

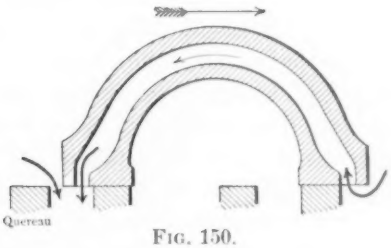

FIG. 150.

Those who oppose its use claim there is no difficulty in getting all the steam into the cylinders that is needed without the Allen port, the main difficulty being to reduce back pressure and compression; that if the Allen valve gives a freer admission, its use necessarily increases the difficulty of getting the steam out of the cylinders, increasing back pressure and compression and causing the engines to pound badly.

Objections.

Strictly speaking, compression is the period of the stroke from exhaust closure to lead opening. The term "excessive compression" is commonly used to denote a suddenly rising pressure at the end of the return stroke which takes the form of a loop on

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

the indicator card, and is usually wrongly assumed to be caused by the compression of steam remaining in the cylinder at the time of exhaust closure. If exhaust closure takes place at half stroke with 3 pounds back pressure and 8 per cent. cylinder clearance, the pressure in the cylinder at the end of the stroke will reach only 113 pounds. If the back pressure is 12 pounds and compression begins at 15 inches, the full stroke being 24 inches, the highest pressure attained at the end of the stroke will be but 137 pounds. With the usual locomotive valve gear and the valve line and line inside, compression begins at about 15 inches of the return stroke. These facts remain true whether the valve is of the common type or an Allen valve. From the foregoing it is quite evident that the Allen valve cannot produce a compression equal to the boiler pressures commonly used, and we must look elsewhere for the cause of the so-called excessive compression complained of. That this cause is excessive lead I believe will be clearly shown by the cards in Fig. 151, which were all taken from the same engine, having Allen valves, under practically the same conditions except a difference in lead, and which clearly prove that the Allen valve can be so set as not to produce excessive compression. (The engine from which these cards were taken had drivers 68 inches in diameter, valves with 5-inch travel in full gear, $\frac{7}{8}$-inch outside lap, and Allen ports $\frac{1}{2}$ by $17\frac{1}{4}$ inches, inside clearance $\frac{1}{16}$ inch on each end, cylinders 19 by 24 inches, and steam pressure 165 pounds.)

In view of the preceding, it seems to me clear that those who criticise the Allen valve have made the mistake of assuming that the excessive compression frequently found to accompany its use is due to so free an admission of steam from the beginning of the stroke up to the cut-off as to produce abnormal back pressure and compression, while the facts in the case are that the objectionable results are caused by too much lead, which can be easily remedied, without losing the advantage claimed for this valve. That the objectionable feature of undue compression can be obviated seems evident from card 9 of Fig. 151.

On card 9, Fig. 151, the average back pressure from the beginning of the return stroke up to the point of exhaust closure is 7 pounds. Taking into consideration the fact that the drivers were making 267.6 revolutions per minute when the card was taken, and that these are representative cards among hundreds taken

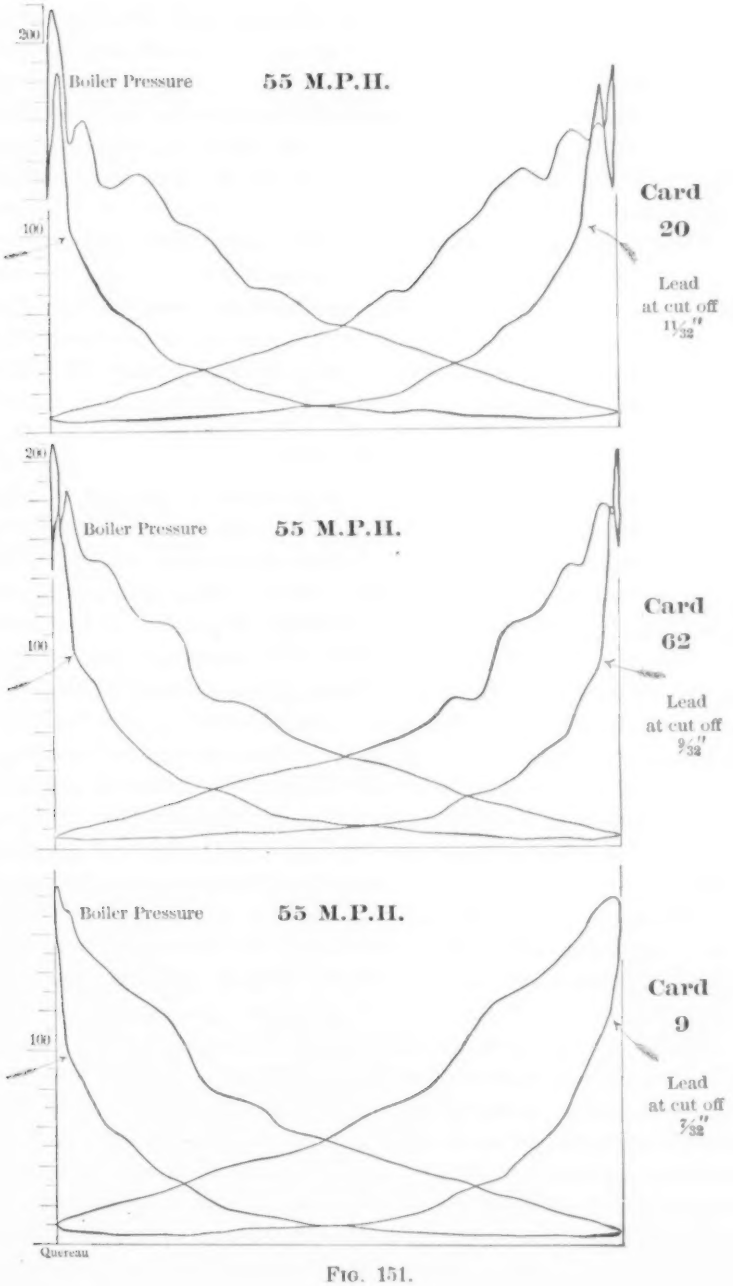
200
Boiler Pressure
55 M.P.H.
100
Card
20
Lead
at cut off
11/32"
200
Boiler Pressure
55 M.P.H.
100
Card
62
Lead
at cut off
9/32"
Boiler Pressure
55 M.P.H.
100
Card
9
Lead
at cut off
3/32"
Quereau

FIG. 151.

under similar conditions, it seems evident that the Allen valve cannot justly be considered as producing more back pressure than the plain valve.

From the foregoing I feel warranted in drawing the conclusion that the Allen valve, when properly set, does not produce more back pressure or compression than a plain valve under similar conditions.

It is sometimes objected that the Allen port necessitates a weak design, or the valves must be made very heavy. For about ten years the Chicago, Burlington & Quincy Railroad has been using, very successfully, on 19 by 24-inch engines, having steam ports $1\frac{1}{2}$ by $17\frac{1}{4}$ inches, Allen valves weighing only 112 pounds without packing strips.

Advantages.

It is evident that the Allen valve doubles the speed of the steam port in opening and closing for all cut-offs, and in this respect more nearly approaches the advantages of the Corliss valve gear than does the ordinary valve. That this is a decided advantage can hardly be successfully disputed. It is equally evident that for all cut-offs, where the maximum port opening does not exceed the width of the supplementary port, the Allen valve doubles the port opening. As the width of the Allen port is quite commonly half an inch, and a maximum port opening of this amount corresponds to a cut-off of 9 inches with the usual locomotive gear, it follows that the Allen port doubles the steam opening for this and all shorter cut-offs, or for all those commonly used for speeds higher than ten or fifteen miles an hour. It therefore appears evident, on theoretical grounds, that the Allen valve increases the power of a locomotive for all speeds above very moderate ones, and that this advantage will be marked for high speeds.

That these theoretical advantages are borne out by practical results is shown by the following quotations from the *Proceedings* of the Western Railway Club for March, 1897: Mr. E. M. Herr, then Superintendent of Motive Power of the Northern Pacific Railway, spoke as follows: "The Allen valve, in my opinion, is a very valuable device if rightly used, and I believe that many railroad men condemn it because they have not used it in the right manner. In making some tests on the Northwestern testing plant, not very long ago, we showed conclusively that at high

speeds a 16 by 24-inch engine would develop more power with an Allen valve than a 17 by 24-inch engine, with practically the same size drivers, would develop with a plain valve. In fact, a 16 by 24-inch engine, on a certain division where the ruling grade could be approached on a good run, was put in freight service with 17-inch engines, and did satisfactory work with the 17-inch engines, pulling over rugged parts of the division, until one day the train happened to stop at the foot of this ruling grade. That day the engine stalled, and it stalled simply because it was not as strong as a 17 by 24-inch engine when pulling at slow speeds. At very high speeds it was stronger."

The following is quoted from the remarks of Mr. Jos. Cockfield, Master Mechanic of the Chicago & Northwestern Railway: "I wish to mention, however, that the 18-inch engine had Allen-Richardson valves, while the 19 by 24-inch engine had plain valves, which, in a measure, accounted for the better performance of the 18 by 24-inch engine."

Writing in 1893, Herr Von Borries, Superintendent of Motive Power of the Hanover State Railways of Germany, said: "The latter (viz., Allen valves) are *most* useful for all fast-running engines. Our people could not do without them."

In the *Proceedings* of the American Railway Master Mechanics' Association for 1896 will be found the report of their Committee on Slide Valves, from which I quote. "From this [table] it will be seen, as might have been expected, that the mean effective pressure with the Allen port is greater than without it at the ordinary running positions. The average value of all the figures shows 20 per cent. greater mean effective pressure in favor of this valve. It can be considered that the Allen port has enabled the piston to exert 20 per cent. more useful pressure on the crosshead at each stroke. It is also readily apparent that the earlier the cut-off the more is gained by the Allen valves. This is also true in a measure of the increase in speed. It is also clearly demonstrated that a better steam line is obtained." The indicator cards on which these conclusions of the committee were based were made on the Purdue University mounted locomotive, at six different cut-offs, varying from full gear to one-sixth stroke, and at speeds of 10, 20, 30, 40, 50, and 60 miles an hour, with the same valve gear and setting for both the plain and Allen valves.

For engines which are already built it seems plain that the Allen valve furnishes a simple method of increasing their power,

and in designing new engines will produce a given power with smaller and lighter valves, because of the shorter ports in the valve seat necessary with this valve, resulting in less strain and wear on the valve gear and seats. It may be urged that a plain valve with longer travel will give as great a mean effective pressure as the Allen valve, but this necessitates a heavier valve and consequent wear to the valve gear, and more power wasted. In this connection it should not be forgotten that the Allen valve enlarges the port opening by a given number of square inches for all cut-offs, while a longer valve travel enlarges it a certain per cent., the number of inches decreasing as the cut-off becomes shorter.

Economy.

Assuming it to be proved that the Allen valve will give greater power than a plain valve, it does not necessarily follow that the economy of the engine will show a corresponding improvement. This will depend on the conditions under which the engine is working. If with the plain valve the conditions are such that the work is done at the most economical point of cut-off, we would not expect any saving in coal by increasing the power; possibly the reverse if the basis of comparison is the pounds of water per indicated horse-power-hour. But if the basis is the cost per ton-mile there can be little doubt that the greater power will secure a saving. That a fuel economy may be the result is shown by Mr. Herr's remarks before the Western Railway Club: "I have endorsed the Allen valve pretty strongly, and I want to say that my experience is just the reverse of Mr. Barr's. The engines in which we put the Allen valve showed quite an apparent fuel economy."

Lead.

Inasmuch as the Allen valve gives a quicker and larger steam admission than the plain valve, it follows that the lead for the former should be less than for the latter. In my opinion it has been the failure to appreciate this fact which has caused many to conclude that the Allen valve has no advantages, but rather the contrary. Not only should the lead for the Allen valve be less than for the plain valve, but it must be if the best results are to be obtained. The smaller lead which the Allen valve requires is a decided advantage in several directions, as will be seen by a study

of the following table, taken from Zeuner valve diagrams of a gear having $\frac{7}{8}$ inch outside lap, $\frac{1}{4}$ inch inside clearance, $48\frac{3}{8}$ inches link radius, 5 inches travel in full gear, and is for 6-inch cut-off:

	Plain Valve.	Allen Valve.
Lead	$\frac{1}{4}$ inch	$\frac{5}{32}$ inch
Lead opens at	$22\frac{7}{8}$ inches	$23\frac{1}{2}$ inches
Maximum port opening	$\frac{39}{1000}$ inch	$\frac{24}{1000}+\frac{15}{1000}=\frac{39}{1000}$ inch
Release begins at	$13\frac{1}{8}$ inches	$13\frac{7}{8}$ inches
Compression begins at	18 "	$18\frac{3}{4}$ "

If we assume that the Allen port is half as efficient as the port opening at the end of the valve, the Allen valve has increased the steam port opening 20 per cent. even after the lead has been reduced nearly 38 per cent. The reduction in lead made possible by the use of the Allen valve has delayed exhaust opening; thus, increasing the period of expansion $\frac{3}{4}$ inch, or 3 per cent., has also delayed exhaust closure $\frac{3}{4}$ inch, or 3 per cent. These results due to the reduction in lead are equivalent to reducing the inside clearance from $\frac{1}{4}$ inch to $\frac{5}{32}$ inch so far as expansion is concerned, and increasing the inside clearance from $\frac{1}{4}$ inch to $\frac{5}{16}$ inch so far as the effect on compression is concerned. There can be no reasonable doubt that these changes introduced by the Allen valve will increase the power of an engine except at very slow speeds, and will favorably affect its economy except when the engine with plain valves is working under the most favorable conditions, which are rare in railroad service.

Indicator Cards.

It may be of interest to some to know that cards taken from Allen valves on single-expansion engines can be distinguished from those taken when plain valves are used. Those from Allen valves have a wavy compression line, such as may be seen on the cards in Fig. 151, while on those taken from plain valves the compression line is a simple curve. The cause for the compound curves on the Allen valve cards I believe to be explained as follows: Fig. 152 shows an Allen valve and seat. Cut-off has just taken place at steam port *B*, and the supplementary port is filled with steam at nearly boiler pressure, perhaps 170 pounds. At *A* exhaust closure is about to take place, when the port and that end of the cylinder will be filled with steam at exhaust pressure, say 10 pounds. As soon as the valve has moved a little farther to the

right the supplementary port will be brought into communication with port *A*, when the high-pressure steam in the supplementary port will expand through *A*, increasing the pressure there and giving the indicator piston a sudden impulse which will cause it

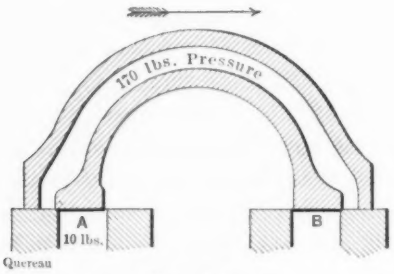

FIG. 152.

to vibrate till lead opens. This explanation is substantiated by the fact that the greater the cubical contents of the Allen port, the more pronounced are the reverse curves.

DISCUSSION.

Mr. Charles T. Porter.—The admirable paper of Mr. Quereau on the Allen valve well illustrates the value of the indicator when used as a means of investigation.

I wish we could have had also, for comparison, diagrams from cylinders with single opening valves, *taken under the same conditions*. But there's the rub. No one except the operator, and he not always, can know that all the conditions—of steam pressure, train resistance, speed, gradient, and uniform motion—*were* the same.

However, the advantage, when cutting off early at high speed, of practically doubling the length of the port for admission is sufficiently obvious, while no one can compare the valve motions given by the eccentric for admission and for release without seeing that it is the former, and not the latter, which become contracted when cutting off early, and so require the double opening.

Perhaps some interest may attach to the early history of Mr. Allen's invention. At the London Exhibition of 1862 I showed, among other things, a sectional model of this valve, working on its seat, beautifully made in mahogany, and nobody paid the least attention to it. During a six years' subsequent residence in

England I tried in vain to get locomotive designers to consider it. Finally, Mr. Aveling was induced to put the valve on one of his road-traction engines. Its operation was entirely successful, and the power of the machine was considerably increased, so that I confidently looked for its adoption by him. After a while I visited his shop in Rochester, to see how it was getting on. I found the valve which had been tried in the store-room, and the project of using it abandoned. Mr. Aveling told me that some of the most eminent engineers in England, who had been consulted in the matter, were agreed that the valve-face would wear unequally, the narrow portions wearing faster than the broad portions, so that it would soon come to leak, and its adoption would ruin his business. He added that he did not believe it, but he was in a position where he could not act contrary to this advice.

After this success in England, I tried what would be done with the valve at home. Through the interest of Mr. Forney, Mr. Buchanan concluded to try it on an express locomotive on the Hudson River Road. He told me that he was just fitting a locomotive with new cylinders, that it would be ready in about a fortnight, that he would like to have me go up with them on it and indicate the cylinders, and would send me word when it was ready. I am waiting for that "word" yet. This was in the reign of the old Commodore, and I concluded that he had forbidden it; he sometimes did such things. A correspondence with the Pennsylvania Railroad Company led to no result. That is all that was ever realized out of the patents, American and English, on this valve, which is now in common use the world over.

It would not, however, have come into general use but for Mr. Forney. He described and illustrated it in his *Catechism of the Locomotive*, and told me that it was the only patented feature that was admitted into that book. This publication brought it favorably to the notice of locomotive designers everywhere.

I cannot refrain from saying, in conclusion, that the consciousness of having been primarily instrumental in giving to the world this useful invention affords me higher satisfaction than I could have derived from any merely pecuniary recompense.

Mr. W. S. Rogers.—In addition to what Mr. Quereau has said in his excellent paper on the Allen valve, I wish to say that locomotives equipped with Allen valves, and running on accommodation train service, making many stops, can pick up their trains much quicker than when using plain valves. They are also quicker in

action, and the coal gate remains in its place in the tank many miles further than is the case with the plain valves.

Practical railway master mechanics were quick to see the value and importance of the Allen valve, but owing to the excessive wear on both valve and valve seat, causing enormous maintenance and expense, although they were made of solid bronze and other composition metals, it was practically abandoned until the late F. W. Richardson applied the Richardson balance, eliminating the excessive frictional load, and gave us the ideal valve action.

I know of wood-burning engines running four years equipped with Allen valves with the Richardson balance, and coal-burning engines running two years without having their valves and seats faced, without which device they could not last thirty days.

DCCCX.*

TEST OF A STEAM SEPARATOR.

BY F. L. EMORY, MORGANTOWN, W. VA.

(Member of the Society.)

THE test recorded below is that of a baffle-plate pattern separator designed to serve as a live-steam separator, as an exhaust-steam separator or trap, and as an oil extractor.

The method of testing the separator consisted, in brief, of a determination of the character or quality of the steam before entering the separator and also its quality after leaving it. Various pressures of steam and different velocities and amounts of steam running through the separator were noted, as also the quality of the entering steam was changed for various runs, as will appear from the readings and quality results of the wet-steam calorimeter.

Apparatus used:

In the first eleven runs noted in the tabulation the following method was pursued: Steam coming from the boiler through a valve A, Fig. 153, passed downward through piping B, which was surrounded by a water jacket, then after two short right bends travelled vertically, giving up, in its vertical portion, a sample to a separating calorimeter CD; with one right bend again, steam after a short connection entered the horizontal separator G. Passing out of the separator the discharged steam gave up a sample to another separating calorimeter M on its way to the condenser, a surface condenser through which the steam passed, as water into a barrel to be weighed. In the last five runs, Nos. 14 to 18, the condenser was not used, but the flow of steam was measured by Napier's law; the steam flowing at higher pressures than before and at higher velocities through a measured nozzle.

In the runs after No. 11 the throttling calorimeter was used in addition, but did not prove satisfactory, the steam being quite too wet for accurate determination.

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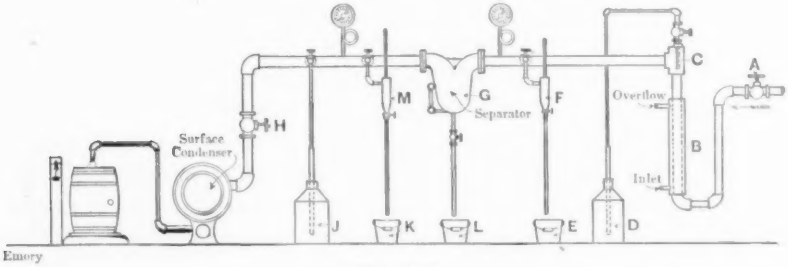
* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

Description of Apparatus.

A is the valve connecting steam main with boiler, through which steam passes along the pipe and thus upwards. At *B* is a cold-water jacket by means of which the steam to be used in the separator test was changed in quality. At *C* a steam nozzle took steam sample from the vertical pipe to a separating calorimeter *D*, and the steam's quality was thus known before passing into the separator. The steam gauge and throttling calorimeter



Emory

FIG. 153.

F also checked this determination. The steam then passed through the separator *G*, and the drips were collected at *L*, the water level in the separator being kept constant. After passing the separator the steam was again sampled by the throttling calorimeter *M* and the separating calorimeter *J*, and passed through the valve *H*, by means of which the velocity was controlled and changed on different runs. The steam condensed by the surface condenser was pumped to the barrel on the platform scales and weighed. The throttling calorimeter on the right was not used with the very wet steam. By using the valves of the drips *L* and *K*, and also the steam used by the calorimeter *J*, the separator itself was used as a check calorimeter. All apparatus was carefully calibrated before the runs.

LOG OF TEST.

Run.	Time, Minutes.	Pressure in Dry-steam Gauge above Atmosphere.	Pressure in Wet-steam Gauge above Atmosphere.	Water from Separator, Lbs.	Condenser Steam, Lbs. Net.	WET-STEAM CALORIMETER.		DRY-STEAM CALORIMETER.	
						Water in Scale, Lbs.	Water in Can, Lbs.	Water in Scale, Lbs.	Water in Can, Lbs.
1	12½	20¾	19.4	22.8	302.95	.023	2.5	Not taken	
2	Not tak'n	34.8	34.3	37.75	307.5	.023	1.9	.045	1.51
3	11½	21.2	20	25	231.87	.16	2.05	.075	1.8
4	7	29.5	27.5	9	335.5	.07	1.4	.058	1.6
5	8	35.5	34	15.5	317.25	.1	1.6	.115	2.15
6	7	11.4	10.6	10	319.5	.08	1.15	.11	1.1
7	8½	16.5	15.8	12	367.25	.08	1.75	.06	1.45
8	6	29.8	29	10	343	.14	1.7	.115	1.65
9	8	86.1	86.6	26	322.75	.215	1.3	.059	1.7
10	9	83.1	83.6	33	365.5	.21	1.15	.09	2.45
11	12	1.3	1	12	355	.04	1.05	.057	.30

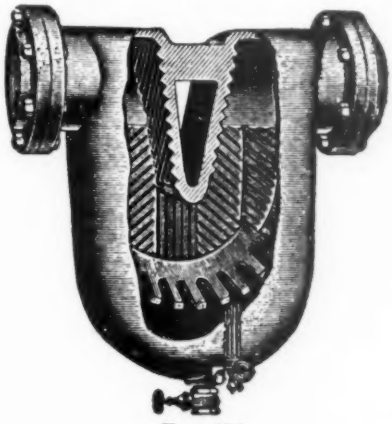


FIG. 154.

LOG OF TEST (*Continued*).

Run.	Time, Mins.	Pressure in Separator in Lbs. Gauge.	SEPARATING CALORIMETER.		Throttling Calorimeter Temp.	Water from Separator, Lbs.	Condensed Steam Used, Lbs.
			Water in Can, Lb.	Water in Scale, Lb.			
12	19	3	.8	.02	212°	9	300
13	9	3.25	.35	.01	216°	16	154

Radiation at 3 pounds = 2.2 pounds for 8 minutes, = .27 pound per minute.
Oil separation: Oil put in 1.42 pounds, taken out .48 pound, per cent. regained = 32 per cent.

LOG OF TEST WITH NOZZLE.

Run.	Time, Mins.	Pressure, Lbs. Gauge.	Separating Calorimeter.		Water from Separator. Lbs.
			Water in Can, Lbs.	Water in Scale, Lbs.	
14	9¼	60	Not taken	—	3.25
15	9¼	47.5	.65	.04	4 9
16	9¼	52	.65	.055	4.72
17	9	52.3	—	—	4.65
18	9¼	70.2	.9	.035	4.95

Radiation = 1.07 pounds in 10 minutes, = .107 pound per minute.
Pressure after leaving or at nozzle averages 1.5 pounds gauge.

QUALITY OF STEAM.

Run, No.	Before Entering Separator.	After Passing Separator.
1		
2	89.2 per cent.	97.1
3	93.2 "	96.1
4	95.2 "	96.5
5	94.1 "	96
6	93.5 "	91
7	95.7 "	96
8	92.4 "	93.2
9	85.4 "	96.6
10	85 "	96.4
11	96 3 "	97
12	97.5 "	99.7
13	95 "	——
14	—— "	——
15	94.2 "	95.3
16	92.2 "	93.3
17	96.2 "	97.3
18		

By using the separator itself as a calorimeter we may find the per cent. of moisture removed by separator. We have by the wet-steam calorimeter the quality of the steam at entering, which, added to the moisture removed by separator, will be the quality after passing the latter. For this experiment, as stated, the nozzle was used, and, if W = the amount in pounds per second flow, F = the area orifice in square inches, and p_1 = the initial pressure absolute, then where $p_1 > \frac{5}{3}$ atmosphere, or p_2,

$$W = \frac{Fp_1}{70}.$$

Here $F = .7854$.

Computing the same for various pressures and runs we have:

Run, No.	Value of W in Pounds Total.	Per Cent. Removed.
14	528	.6
15	408	1.1
16	416	1.1
17	406	1.1
18	530	.9

These per cents. added to the quality before entering give the values shown for runs 14 to 18.

From the general run of the results as shown on the table of qualities it would seem that runs 6 and 12 should be thrown out as being in all probability associated with some error of conditions or observations. If we do so we would conclude that this separator in good condition and for the higher velocities of steam flow will deliver steam of, say, 97 per cent. dryness, and has little separating qualities in steam of dryer quality. And we are quite safe in concluding that it will, if kept well drained, deliver a steam dry within 3 per cent. There seems to be little difference in its action for live steam from boiler and the exhaust steam or steam of low pressure of considerable wetness. The steam of 3 pounds above atmosphere, corresponding to ordinary engine exhaust, was, as will be seen from its quality in the table, made to contain by means of the jacket considerable moisture, and again made quite dry. The action, however, was similar to that for steam of higher pressure and similar wetness, namely, to deliver steam of, say, 97 per cent., irrespective of original quality.

Test of a Baffle-plate Steam Separator as an Oil Separator.

Test of the above separator as an exhaust-steam separator and oil separator.

The apparatus before mentioned was used in the test as an oil separator; the calorimeters and their leading pipes and all unnecessary attachments were removed, and a force pump, small pump oiler, was attached at the bottom of the first vertical pipe leading to the separator as described before. A steam gauge was attached at a point beyond the separator, which recorded the pressure in the separator, and the steam was allowed to flow through and into the air, discharging as in ordinary practice in a non-condensing engine.

The method pursued was, in brief, to pump a known amount of

cylinder oil into the pipe leading to the separator, and to draw away from the separator through the drip at the bottom the oil and water separated from the steam on its passage. The per cent. of efficiency would evidently be the ratio of the amount taken out to the amount put in, multiplied by 100.

This was done: a goodly supply of "vacuum" cylinder oil was pumped into the pipe slowly as it would be in working conditions in the engine manipulations, and the oil coming away with the water was carefully skimmed off and weighed. Much care was exercised in the beginning to see that a quantity of oil sufficient to oil all the parts between the pump and separator was put in so that all interstices and joints might absorb or take up all that they might hold and so to insure that no oil would be caught and held before entering the separator and after leaving the pump. In fact a run was made at first, which is not here recorded, which served as a zero point to begin observations.

It was found that it was impossible to skim the mixture of oil and water so carefully as to warrant saying that the skim was only oil, although to the eye there were no water globules present, but the specific gravity of the oil regained was much heavier than that put in, and the weight of mixture taken out was much greater than that of oil put in. The mixture was cooled to 60 degrees Fahr., and from its weight and specific gravity was figured the probable weight of oil in the mixture. The water which existed in the oil or mixture is held in very small globules between the parts of the heavier oil, and therefore affects the specific gravity and weight directly in the proportion of its existence. This mixture being homogeneous, we have the following method of calculating the oil present:

Let x = weight water in mixture.

Let y = weight oil in mixture.

Let V_1 = volume water in mixture.

Let V_2 = volume oil in mixture.

Let $V_1 + V_2$ = volume oil and water in mixture = V_3.

Then the relative volumes equal the weight divided by the specific gravity, or

$$V_1 = \frac{x}{1},\ V_2 = \frac{4}{.893},\ V_3 = \frac{5.1}{.942};$$

then

$$x + \frac{y}{.893} = \frac{5.1}{.942}\text{; but, again,}$$

$x + y = 5.1$, $x = 5.1 - y$,

$$5.1 - y + \frac{y}{.893} = \frac{5.1}{.942},$$

$$y\left(\frac{1}{.839} - 1\right) = \left(\frac{5.1}{.942} - 5.1\right),$$

$$y = \frac{\frac{5.1}{.942} - 5.1}{\left(\frac{1}{.891} - 1\right)} = \frac{.314}{.19418} = 1.6176 \text{ pounds.}$$

But amount put in = 3.05, and since amount taken out is 1.6176 then efficiency $= \frac{\text{Oil taken out}}{\text{Oil put in}} = \frac{1.6176}{3.05} = 53$ per cent.

TESTS.

Run.	Oil Put In.	Mixture Taken Out.	Specific Gravity Mixture.	Per Cent. Oil Regained.
1	1.86 pounds	2.81	.930	58.6
2	3.05 "	5.1	.942	53
3	1.4 "	1.82	.940	42
Average				51.2

The specific gravity of the oil put into the pipe through pump was .893 at 60 degrees Fahr.

These experiments prove that, in round numbers, 52 per cent. of the oil was regained, although the mixture which might be used for lubrication again was a much larger amount.

DISCUSSION.

Mr. Francis H. Boyer.—For steam users who are situated where they have to use salt water this becomes an important feature. When you think for a moment that we have to consume all the way from 9 to 15 per cent. of the valuation of the fuel consumed for the water that we use, it becomes a consideration of which, when I read the title of this paper, I was in hopes that Mr. Emory would give us some solution, but we find none.

I would not question for a moment the first clause of his paper, where he speaks of the separation of the water, but, taking up the steam separation, probably there is not a manufacturer or engineer in the room but what is intensely interested in this question of separating oil from exhaust steam, and on shipboard you know what that means. The oil accumulates, and let ever so small an amount of it get into the water, you have serious trouble with your boilers. I have a plant where, with a series of engines, we run about 800 horse-power into one exhaust line, and to extract the oil from the exhaust steam, putting in one of these large baffle-plate separators, we caught this oil, and convert it and re-use it for lubricating purposes. If I had been in the packing business I could have made better use of that oil for packing purposes than lubricating. It came out in a spongy mass, after being heated to the temperature of the steam—312 or 320 degrees—it became a spongy mass with no lubricating properties. There was nothing of oil left, and it was absolutely of no value for lubricating purposes. There is a device known to men in the ice-manufacturing business for separating oil from exhaust steam that does its work perfectly. They build a large cylinder or tank, putting a baffle-plate in the bottom; as the steam comes up, distributing it, and then letting it pass through any substance by which this oil will become coated on the substance it may contain. I found one cylinder filled with bluestone spawls; another one with brick. I saw another one filled with wood; another one that I knew of where they put salt grass. But in my own work I would prefer to use with that class of separator coke or charcoal, for the reason that after this substance becomes once covered with oil you can burn it, and by using coke you can put it in your furnace to make fuel. That, in association with a surface condenser, makes a very complete plant. I do not know of one that is being run that way, although I have designed one and contemplate its building at an early date. For 1,000 horse-power I shall build a cylinder 6 feet in diameter to 12 feet in length, about 24 inches on the bottom. We will use for the first operation an oil separator of the baffle-plate design. In this I should expect to get out 70 or 80 per cent. of the oil, then passing it through the coke, and as it comes up through the separator, the time it will take to deposit oil on the surface of the coke—ten days or thirty days or sixty days, whatever it may be—will establish the length of the filling, and when you are through open your manholes and take your coke out and

burn it, and put in a fresh supply. Erecting this in contact with a condenser beyond the separator, we find a loss not over from 2 to 3 inches of the vacuum running at 22 to 23 inches. In Boston, in one of the oldest installations of our large electric plants, where they are running off 4,000 horse, their water costs them $6,000 a year. They are running their water direct through jet condensers and into the river. They did buy a large steam separator of the baffle-plate description, and separated as large a per cent. of oil as possible, and prevent that from running on the surface of the Charles River. But I know not now of a large plant where this separation of oil from the water is sufficient to permit this being used continually, and I do not believe there is a baffle separator that is made that will separate oil from exhaust steam in a manner satisfactory, and for the re-using of the water. I think it is an important question, not so much the question that is brought out by Mr. Emory in his paper, but this question of the separation of oil from water; that it is one that means a good deal of saving and economy, and one in which all engineers are interested.

Mr. Emory.—I would add only a few words in explanation of the paper, and say that no test of the lubricating qualities of the oil separated during the test was made. As the oil did not pass through an engine cylinder, and encountered actual demands of lubrication, it was thought such a test would not be of much value. A chemical test, also a physical test, was intended to be added. But as reported, it was simply a separation test and nothing else. As the separation of the residue is the first essential in reclaiming and cleansing, it does not appear to me, as affecting the efficiency of a separator of this type, whether this residue has lubricating qualities or not. The chemical changes in the oil due to the heat of the engine cylinder or steam cannot justly be charged to the separator. I trust Mr. Boyer will show the efficiency of his contemplated arrangement when completed, together with tests of the lubricating quality and value of the reclaimed oil. I for one shall be much interested.

DCCCXI.*

EXPERIMENTS ON VARIOUS TYPES OF FIRE HYDRANTS.

BY CHAS. L. NEWCOMB, HOLYOKE, MASS.

(Member of the Society.)

THE fire hydrant, called in many places a fire plug, is an important part of the fire apparatus which our cities and towns are providing to guard against the fire hazard. The importance of this apparatus is at once seen when we remember that on the average each year over one hundred million dollars' worth of property is destroyed by fire in the United States.

The object of these tests was a complete investigation of the fire hydrants now commonly used, and the work was divided into the following classes:

1. The loss of pressure due to the friction of water in the hydrant, the total loss being subdivided into barrel loss and nozzle loss.

2. The discharging capacity of open hydrant butts at different pressures.

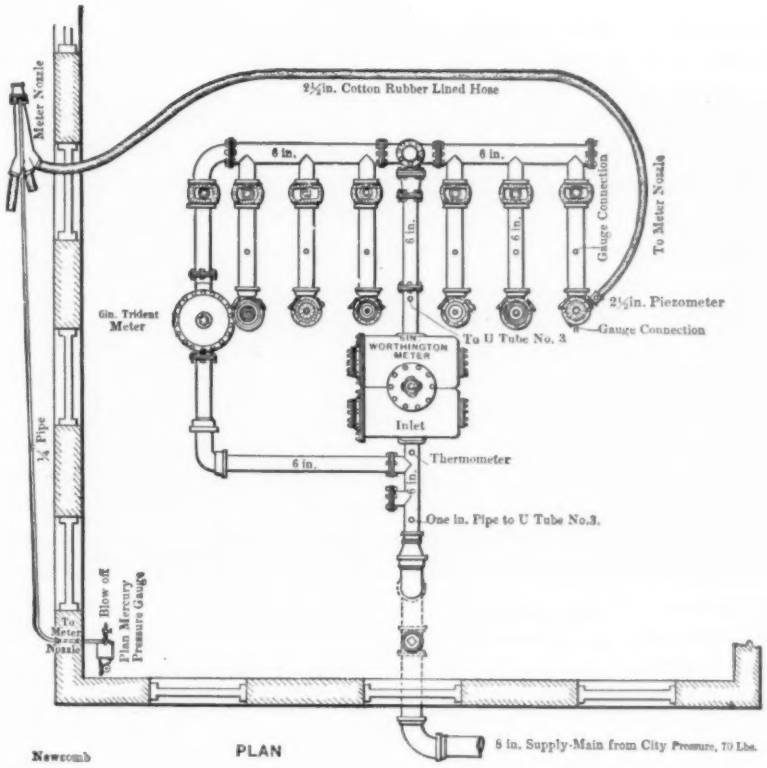
3. The water-hammer caused by quickly closing the main gate.

4. General features of construction, certainty of action, strength, durability, etc.

Some interesting data were also obtained on two 6-inch meters which were used in the tests.

A few tests on several different hydrants were made in 1886 by Prof. Selim H. Peabody, of the University of Illinois. The results of these were not entirely satisfactory. Mr. John R. Freeman, in his "Hydraulics of Fire Streams,"* gives a table of the discharge of one open butt of a four-way independent gate hydrant at various pressures and some percentage corrections when using the table for other types of hydrants. Beyond these tests the writer has been unable to find anything of im-

* Presented at the Washington meeting (May, 1899), of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.



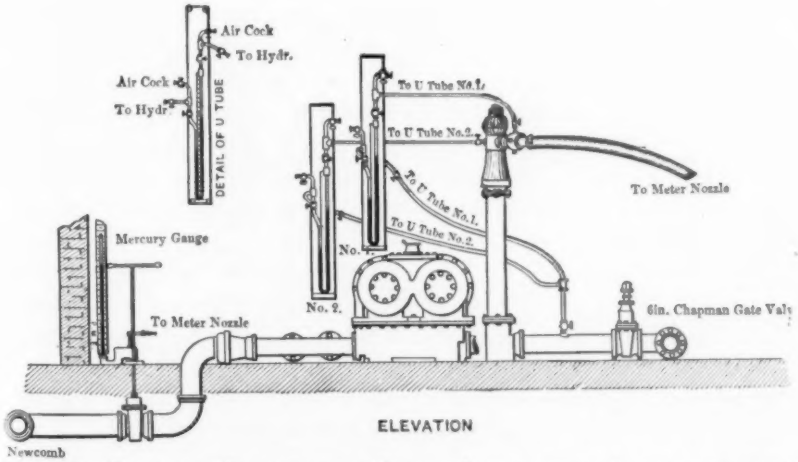


FIG. 155.—PLAN AND ELEVATION OF PIPING, SHOWING THE ARRANGEMENT FOR TESTING OF FIRE HYDRANTS, HOLYOKE, MASS., NOVEMBER, 1897.

portance, though diligent inquiry was made. In the experiments here described it was possible to test a large number of hydrants under widely varying conditions, and the aim has been to bring out results which would be of value to hydraulic engineers and hydrant manufacturers.

The tests were made for the Water Department of the City of Holyoke, Mass., at the request of the Water Commissioners. They were carried on in the basement of the Water-Works shop, commencing in November, 1897, and ending in June, 1898, with an intermission of several months. The full facilities of the Water Department were at all times available, thus making it possible to go into the investigation with much thoroughness.

The writer is indebted to Mr. John R. Freeman, of the Associated Factory Mutual Fire Insurance Companies, for valuable suggestions from his large experience in testing work, and for the general coöperation of the Inspection Department of Factory Mutuals. Mr. Frank L. Pierce and Mr. E. V. French, of that department, made frequent visits to the testing-room during the experiments, and gave much help in planning the scope of the tests and the arrangement and handling of apparatus. The testing work done in 1897 and in January, 1898, was under the immediate charge of Mr. Ezra E. Clark, of Springfield, Mass. The work done in June, 1898, was under the immediate charge of Mr. A. L. Kendall, of the Factory Mutuals. Messrs. French, Clark and Kendall have also aided in carrying on the computations and in getting the report in shape for publication.

The hydrants were in part bought from the manufacturers and in part loaned by them. In all cases it was fully explained that the hydrants were to be used in extensive tests, and that the usual commercial article was desired. The cheerful cooperation of the manufacturers was of material help throughout the experiments.

The mercury gauges and the meter nozzle were loaned by the Inspection Department of the Factory Mutuals. The Trident meter was loaned through the kindness of the Neptune Meter Company. The Worthington meter was purchased specially for the work. The American Thompson Improved Indicators were loaned by the American Steam Gauge Company.

* "Experiments Relating to Hydraulics of Fire Streams," *Trans.* American Society of Civil Engineers, Vol. XXI., Table B-No. 3.

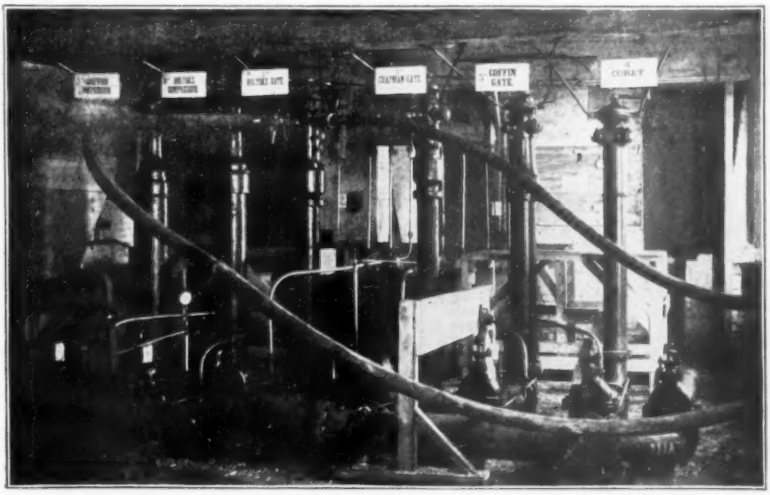


FIG. 156.—INTERIOR OF TESTING-ROOM.

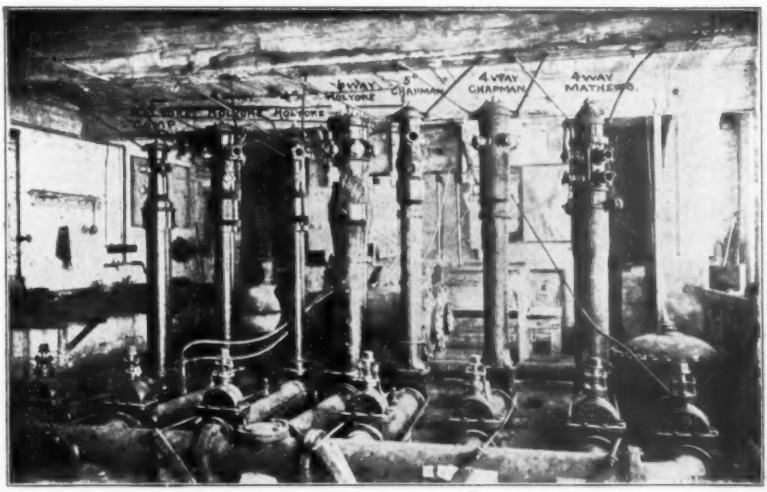


FIG. 157.—INTERIOR OF TESTING-ROOM.

Arrangement of Apparatus.

The water for the tests was drawn from the city mains, an 8-inch pipe being laid into the basement for this work. The city is supplied by gravity from reservoirs, and the pressure was generally quite steady. The mains were of such size that all water needed could be obtained at good pressure. Fig. 155 shows the general arrangement of apparatus. The 8-inch main was provided with a gate just inside the building. It then connected with a 6-inch Worthington meter. A short length of pipe connected the meter to a manifold with six outlets. To each outlet was attached a 6-inch gate, and the hydrants connected with the gate by a short length of 6-inch pipe. Figs. 156, 157, 158, and 159, made from photographs, show the general appearance of the apparatus.

Piezometers.—About 2 feet from the inlet of the hydrant a hole for $\frac{3}{4}$-inch pipe was drilled in the cast-iron pipe and tapped. Into this hole a short nipple with a stop-cock was screwed, from which connections were taken for the gauges. This connection served in place of a regular piezometer and can be relied upon to give sufficiently accurate results, as the velocities were not high and there were several feet of straight pipe back of the hole, so that a uniform condition of flow might be expected. Care was taken to drill the hole at right angles to the pipe and to prevent the gauge connection from extending into the pipe. Care was also taken to remove any burr made in drilling. If either of these points had been neglected a false reading would have been the result.

A $\frac{1}{4}$-inch hole was tapped into the back of the hydrant about on a level with the nozzles, and a connection screwed in with the same care as for the 6-inch pipe. To the $2\frac{1}{2}$-inch hydrant outlets special piezometers, as shown in the sketch on page 500, were attached. The circular space extended around the piezometer and connected with the interior by four $\frac{1}{8}$-inch holes drilled at right angles to the axis of the bore. The pipe leading to the gauges was connected to this space between two of the holes. This arrangement materially assists in the accurate determination of the pressure when the velocity is high and the water is likely to be in eddies, as would be the case at the outlet of a hydrant. At the end of the piezometer a 50-foot line of ordinary $2\frac{1}{2}$-inch cotton, rubber-lined, jacketed fire hose,

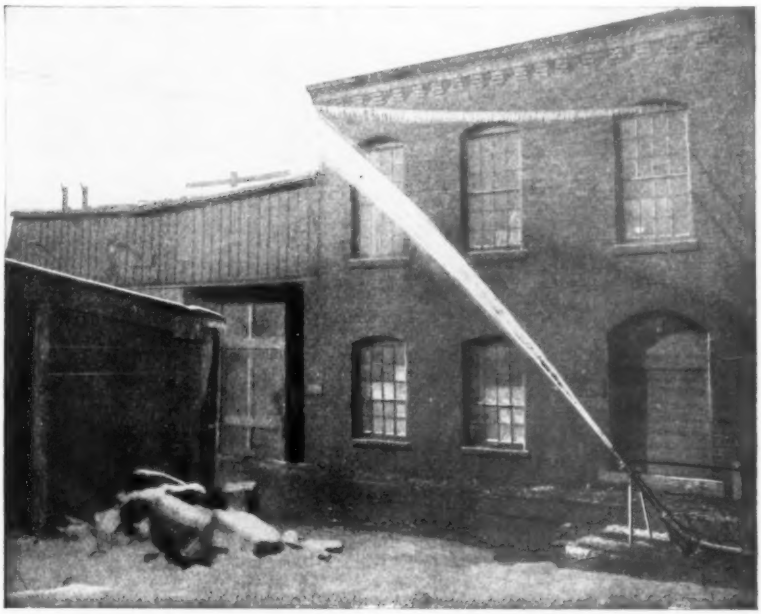


FIG. 158.—EXTERIOR OF TESTING-ROOM, SHOWING METER NOZZLE IN OPERATION.

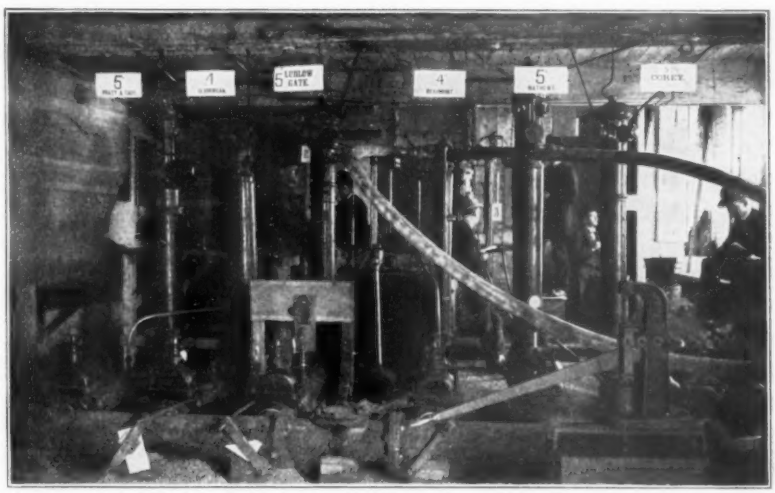


FIG. 159.—INTERIOR OF TESTING-ROOM, SHOWING HYDRAULIC TEST PUMP ON RIGHT AND WATER-HAMMER RIG TO LEFT OF CENTRE.

loaned by the Fire Department, was attached, which conducted the water to a meter nozzle located in the yard. When two outlets were in use two lines of hose were employed, and for the three- or four-way hydrants three and four lines were used. With two lines of hose two piezometers were used and connected together by a short length of $\frac{1}{4}$-inch pipe. The connection for the U-gauge was taken from about the middle of this pipe. When using more than two streams, only two piezometers were used, it being assumed that the average pressure was fairly well secured in this way without the use of a piezometer on each outlet.

Meter Nozzle.—The meter nozzle was one designed by Mr. John R. Freeman for accurate water measurement. A cut of it is shown

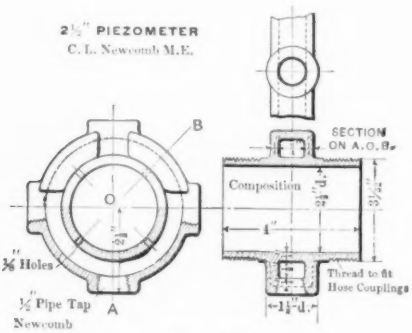


FIG. 160.

in Fig. 161.* The three-way connection joins a $3\frac{3}{4}$-inch smooth-bore play pipe, having at its end a $3\frac{3}{4}$-inch piezometer with gauge connection at C made on the same principle as the $2\frac{1}{2}$-inch piezometer, shown in Fig. 160. To the end of the piezometer nozzles from $\frac{1}{2}$-inch to $2\frac{1}{2}$-inch bore can be screwed. When using four streams an ordinary Siamese connection was screwed to one of the inlets. For the six streams in the tests of the six-way hydrant, the fifth and sixth lines were run separately, and each provided with an Underwriter play pipe. At the base of one play pipe was a $2\frac{1}{2}$-inch piezometer, which was connected to the mercury column and turned on alternately with the meter nozzle.

With the pressure at the nozzle piezometer accurately known the quantity of water can be readily computed. In all the fric-

* A full account of the original tests of this nozzle will be found in the *Transactions* of the American Society of Civil Engineers, Vol. XXIV.

tion loss tests the quantities were determined by the nozzles rather than by the meter. The Worthington meter gave trouble from its pulsations affecting the gauges, so that the by-pass connection shown at the left in Fig. 155 was put in and the meter abandoned. Later the Trident meter was put into this connection and used for the last part of the open-butt tests.

Gauges.—The nozzle pressures were measured by a mercury

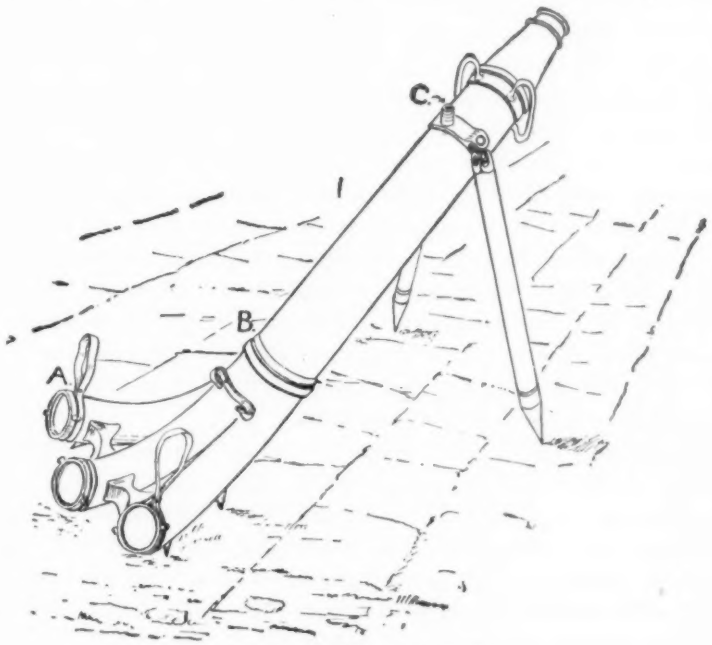


FIG. 161.—METER NOZZLE.

column * with a scale graduated to read pounds and tenths of a pound directly and to include a connection for the lowering of the surface of the mercury in the cistern. The gauge was connected to the nozzle piezometer by ¼-inch iron pipe and rubber tubing, care being taken to have the joints practically tight. The pipe sloped upwards from the nozzle, and a blow-off cock was placed at the highest point, so that all air in the connections could be surely blown out. The gauge was set up against the

* For the general construction of the mercury column, method of graduating the scale to compensate for lowering of the cistern, weight of mercury, etc., see "Hydraulics of Fire Streams," by J. R. Freeman, *Transactions* A. S. C. E., Vol. XXI.

rear brick wall of the building, and the tube extended into the second story of the shop, through a hole in the floor.

Losses of pressure were measured between the 6-inch inlet and the connection at the top of the hydrant, also between the 6-inch inlet and the 2½-inch piezometer. Thus the first gave the barrel loss and the second the total loss. The difference was then the nozzle loss. These losses were measured directly on a U-tube mercury gauge made and connected about as shown in Fig. 155. For these gauges a heavy glass tube with an inside diameter of about ¼ inch was bent into the shape of a U, with one leg about twice the length of the other. The tube was filled with mercury nearly to the top of the shorter leg. A short length of rubber tube was cemented to the top of each leg. This tube connected the glass to a fitting made up of ordinary ¼-inch pipe and containing stop-cocks and a blow-off cock for air. From these fittings connections of ¼-inch iron pipe or of rubber hose were taken to the points on the hydrants between which the loss was desired. Care was always taken to have the connections slope upward from the hydrant to the air-cocks, so that when the cock was opened the swift flowing current of water washed out all air in the connections.

These gauges gave directly the gross loss of pressure between the two points with which connection was made, this loss being the difference in height between the two columns, less a slight correction for a column of water of a height equal to this difference. A measuring stick graduated directly in pounds and tenths of a pound was fitted between the legs of the tube and gave the readings directly in pounds.

Open-butt Tests.—In the open-butt tests the water was discharged directly from the 2½-inch hydrant butt. The floor of the test-room was of brick laid in cement, with a large drain all prepared especially for this work, so that the water could be blown out anywhere in the room.

For the tests through No. 368 the mercury gauge was connected to the 6-inch piezometer at the inlet of the hydrant, and a U-gauge connected between this point and the connection on the back of the hydrant. The pressure at the outlets was then the mercury-column reading corrected for elevation, less the friction loss through the hydrant, which was given by the U-gauge. After test 368 the mercury column was connected directly to the nipple at the back of the hydrant barrel opposite

the outlets. This gave the pressure directly at the outlets, and was better than the first arrangement, as it avoided the errors of the U-gauge.

Method of Conducting Test and Computing Results.

Friction Losses.—In all the tests observers were placed at the different gauges and took readings simultaneously at the sound of a bell, a warning bell being struck five seconds before the time for the reading. The majority of the tests were of ten minutes' duration, and readings were taken each minute. A few of the tests were of five minutes' duration, and in these half-minute readings were taken.

The following programme was adopted for all the friction-loss tests:

Condition.	Size of Meter Nozzle.	Approximate Pressure at Nozzle.	Approximate Gallons per Minute Flowing.
One-hose outlets	¾ inch	66 pounds	130
" "	1⅛ inches	46 "	250
" "	1¼ "	33 "	350
" "	1⅜ "	24 "	450
" "	2 "	19 "	550
Two-hose outlets	1¼ "	65 "	375
" "	1⅜ "	30 "	500
" "	1⅝ "	45 "	625
" "	2 "	34 "	750
" "	2½ "	17 "	850

In all cases after the water was started ample time was allowed before the readings were commenced to make sure the water had come to a steady condition of flow.

Before or after each series of tests the zero reading of the mercury column was obtained by filling the gauge connections with water and holding the end level with the centre of the nozzle and then reading the gauge.

At the end of a series of tests the averages were quickly computed and checked, and the gross loss in pounds, and the gallons per minute discharged for each condition, were plotted on cross-section paper and a curve drawn through the points. This gave a constant check on the work and quickly showed up any error. The occasional prompt finding of a discrepancy tended strongly to impress the observers with the need of care

in taking readings and handling the apparatus. For this reason and for the greater ease with which any trouble is located and remedied, and for the chance of studying the results and investigating at once while the apparatus is in place any special feature, this method of carrying the computations along with the work is believed to be of the greatest benefit to the experimenter. The chance of false readings was carefully guarded against by liberal blowing off of all connection before each test. For a number of hydrants, series of tests were repeated to try the accuracy of the work, and it was almost invariably found that the two series agreed well within practical limits.

The question was raised as to whether the high velocity of water through the 2½-inch hydrant butt might not cause such a contraction of the stream as to affect the readings of the U-tube. Two series of tests were therefore made on one hydrant, one with the 2½-inch piezometer next to the hydrant outlet, which was the usual arrangement, and the other with a piece of 2½-inch pipe about 2 feet long between the hydrant and the piezometer. A special test for the friction loss in this 2½-inch pipe was then made, and, correcting for this loss, the results were found to be practically the same in the one case as the other, showing that the piezometer screwed directly to the hydrant could be relied upon for accurate results.

Computations.—The average readings from each test were copied onto data sheets arranged with parallel columns. The main steps of the computations were also put onto these sheets, thus giving a complete record of the work. The sheets appear at the end of the report. The friction-loss tests are arranged in alphabetical order according to the names of the hydrants, with all the tests on one make of hydrant grouped. The open-butt tests are in order of test numbers. For a few of the hydrants complete tests were not made, for lack of time. The absence of data will show where these omissions occur.

For the friction-loss data sheets the following explanations may be made:

Column 4 gives the size of the nozzle tips used in the different tests.

Column 5 gives in the first part the actual reading of the mercury gauge attached to the nozzle, and in the second the pressure corrected, as already described, for the difference in height between the centre of the nozzle and the zero of the

gauge; that is, the second column gives the actual pressure at the nozzle.

Column 6 is obtained directly from column 5 by curves plotted from the following formula, which is taken from "Hydraulics of Fire Streams":

$$G = \frac{29.83\ cd^2}{\sqrt{1 - c^2\left(\frac{d}{D}\right)^4}}\sqrt{p\,1}$$

d = diameter of orifice of discharge in inches.
D = diameter of piezometer in inches.
c = coefficient.
p = pressure in pounds per square inch at centre of orifice.
The following coefficients were used:

$\frac{3}{4}$-inch nozzle	Coefficient 0.985
$1\frac{1}{8}$ " "	" .99
$1\frac{1}{4}$ " "	" .99
$1\frac{3}{4}$ " "	" .995
2 " "	" .995
$2\frac{1}{2}$ " "	" .995

Column 7 was obtained by an ordinary Bourdon gauge attached to the piezometer connection in the 6-inch pipe supplying the hydrant. These readings were taken to show the approximate pressure worked at and thus make the data complete. The friction losses were of course independent of the pressure.

Columns 8 and 9 show the gross losses read directly from the U-tube gauges, and also these losses corrected for the water column, as already explained.

Column 10 is merely the difference between columns 8 and 9, and is therefore the gross nozzle loss.

Columns 11, 14, and 16 are computed from the quantity flowing as given in column 6, and columns 12, 15, and 17 are obtained from these velocities by the formula $h = \frac{V^2}{2g}$. The losses given in columns 8 and 9 are not actual friction losses, as they contain the gain or loss of head due to changes in the velocity of water. For example, the actual friction loss between the gauge connection on the 6-inch pipe supplying the hydrant and the $2\frac{1}{2}$-inch piezometer on the hydrant outlet, *i.e.*, the total loss (column 19), is the gross loss of column 8 less the loss of head necessary to change the velocity from that in the 6-inch pipe to

that in the 2½-inch pipe. This loss is the difference in the heads corresponding to the two velocities, which is column 17 less column 12. The actual friction loss is therefore

Column 8 — (column 17 — column 12) = column 19.

When two lines of hose were in use half the total quantity was assumed to be passing through each outlet; with three lines one-third each, and four lines one-quarter each.

Column 18 is figured in a similar way.

Column 20 is simply the difference between columns 19 and 18, which is the nozzle loss.

Open-butt Tests.—The general method of procedure was the same as for the friction tests. The mercury column and the U-gauge, in the tests where it was used, were read in the same way and with the same care as in the friction-loss tests. The quantity of water was obtained from the meters, the Worthington meter being used in all of the earlier tests, the Trident meter in the later ones. The exact time in which the meter registered an even number of cubic feet was noted with a stop-watch. To get an even number of cubic feet a definite number of revolutions of one of the dial hands was timed. This method is free from possible inaccuracies in the graduations of the meter registers.

The average readings from each test and the main steps of the computations have been placed on data sheets similar to those used in the friction-loss tests. These sheets are placed at the end of the report, and in them the arrangement is according to test numbers, this being more convenient under the conditions than an alphabetical order. The following explanations may be made.

Column 2 is the duration of test from the stop-watch readings.

Column 7 is the reading of the mercury gauge.

Column 8 is the correction. In the earlier tests this is in two parts—the first part for the difference in elevation between zero of the scale and the centre of the connection at the top of the hydrant barrel. The second part of the correction is the friction loss in the barrel given by the U-tube reading. After the U-tube was abandoned, the correction is simply for elevation.

Column 9 is column 7 less column 8, and is therefore the real pressure at the back of the hydrant outlet.

Column 10 is the total cubic feet passed in the time given in column 2.

Column 11 is computed from columns 2 and 10.

Column 12 is read directly from the meter-calibration curve.

Column 13 comes from column 12, using the assumption that at a given pressure the discharge varies inversely as the square of the diameters.

On several hydrants tests were run, first, with the pressure taken at the usual connection at the back of the barrel, and, second, with the pressures taken at a connection tapped into a cap screwed to the other outlet if a two-way hydrant, and to any one of the other outlets if the hydrant had more than two outlets. The plotted points from the two tests gave practically the same curve, showing that where more convenient there is no objection to using a tapped hydrant cap.

Curves.—In addition to the rough plottings which, as already stated, were made as the work progressed, final curves were drawn from the completed results. It was from these curves, which give the best means of averaging the several experiments on any one hydrant, that the data for the tables designed for practical use were taken.

Plates I. to XXII., inclusive (Figs. 162 to 183), show friction losses, and are columns 18 and 19 of the data sheets, plotted with column 6, thus giving actual friction losses for one coördinate and gallons per minute flowing for the other.

I

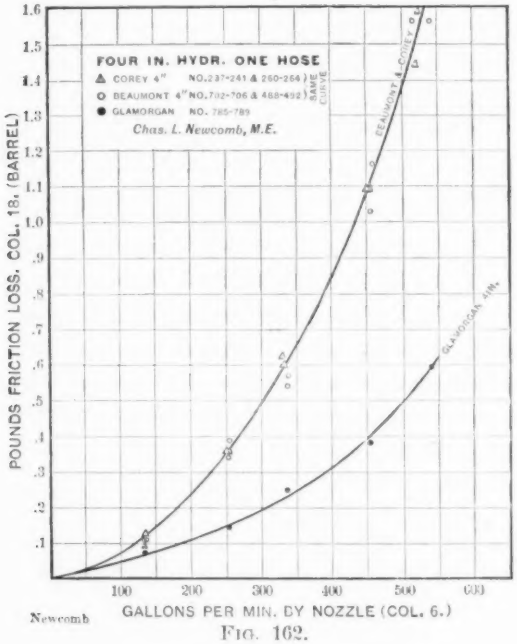


FIG. 162.

II

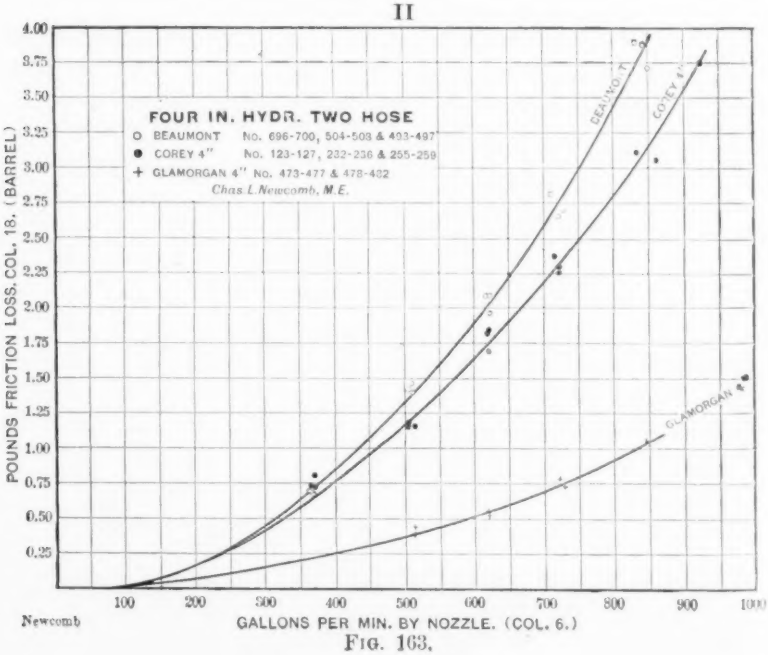


FIG. 163.

III

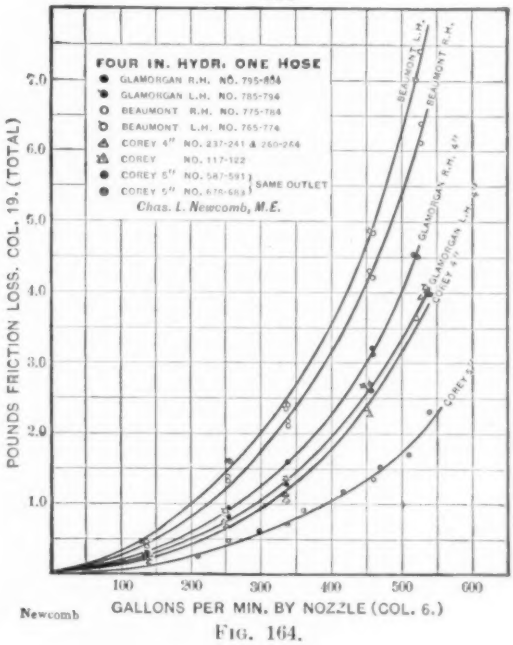


FIG. 164.

IV

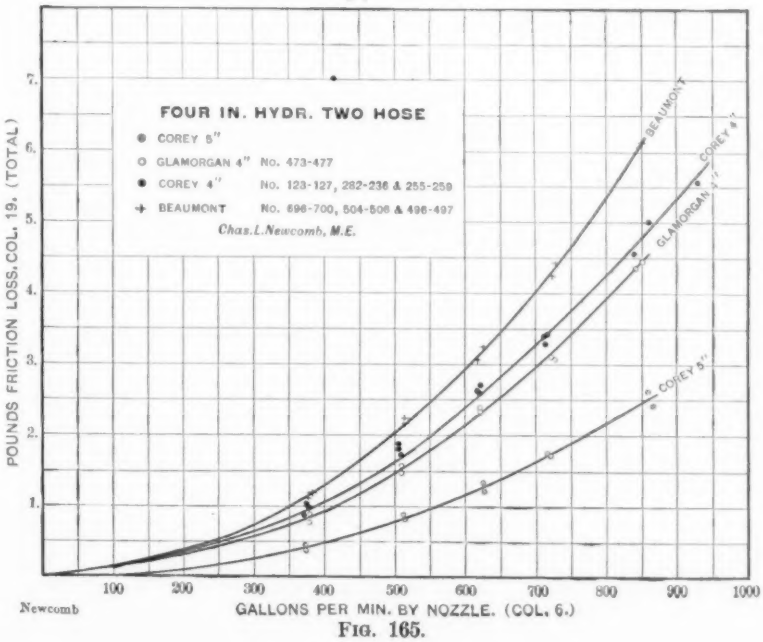


FIG. 165.

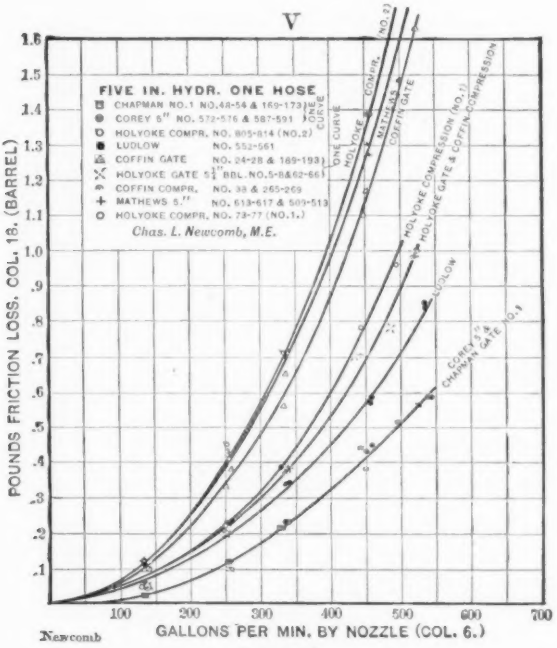


FIG. 166.

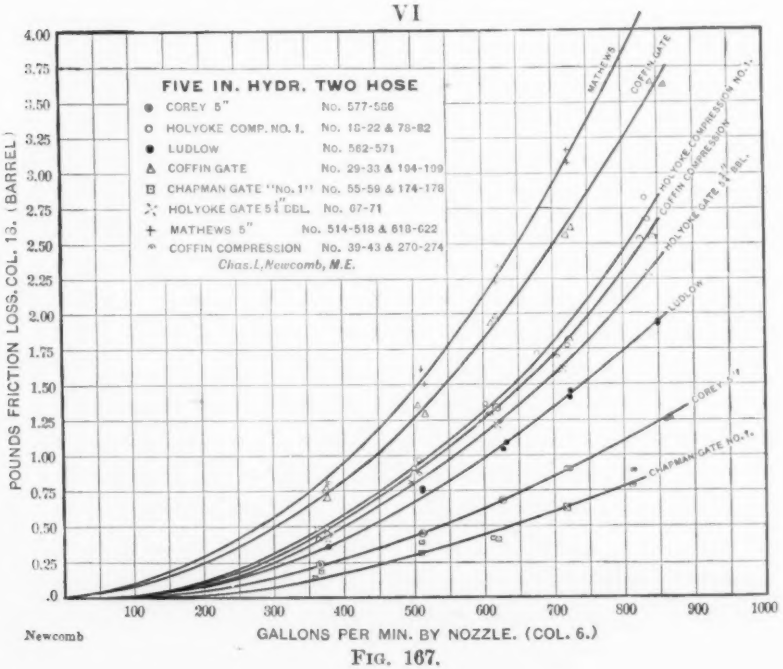


FIG. 167.

VII

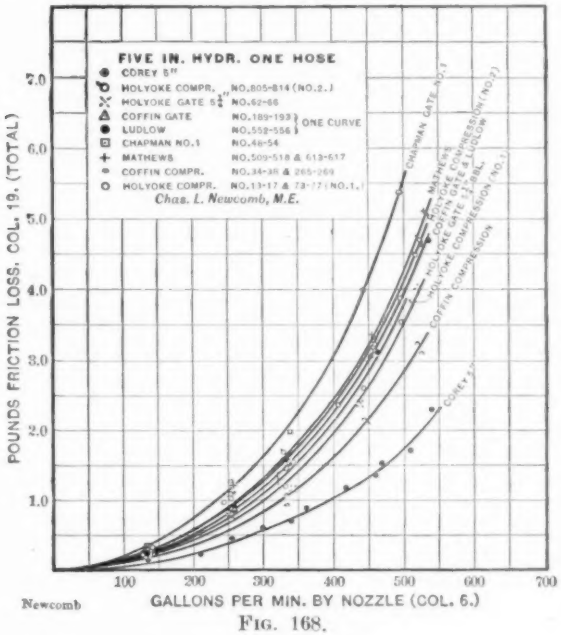


FIG. 168.

VIII

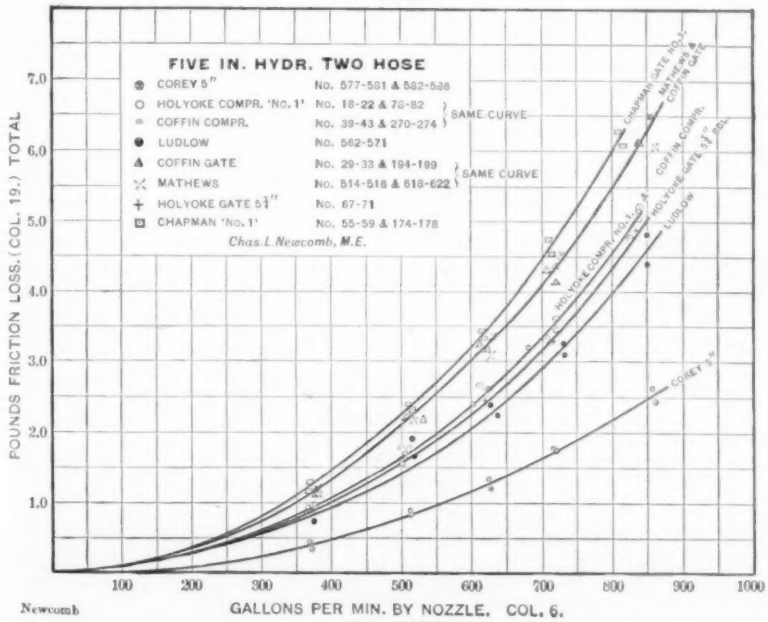


FIG. 169.

IX

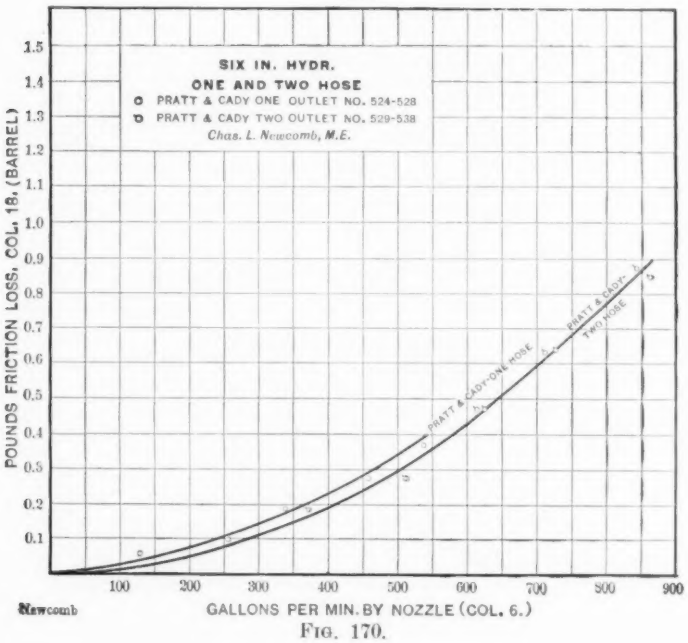


FIG. 170.

X

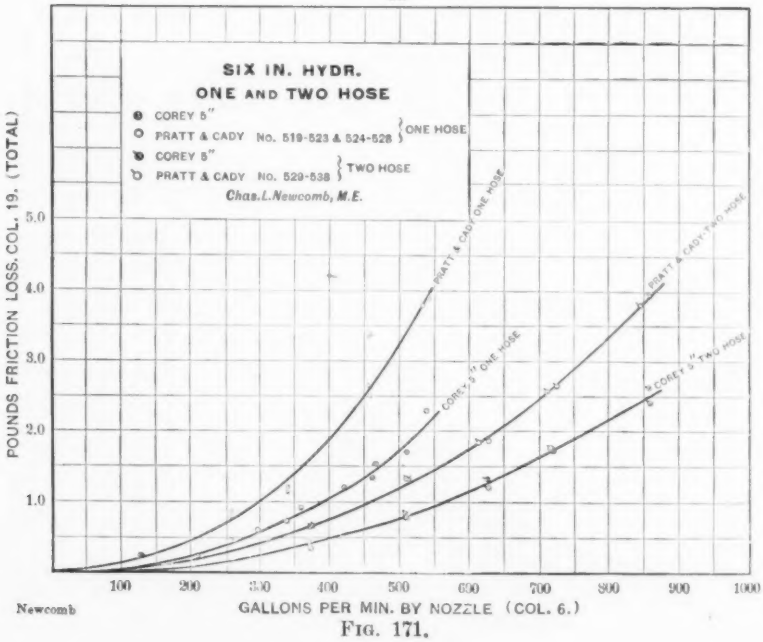


FIG. 171.

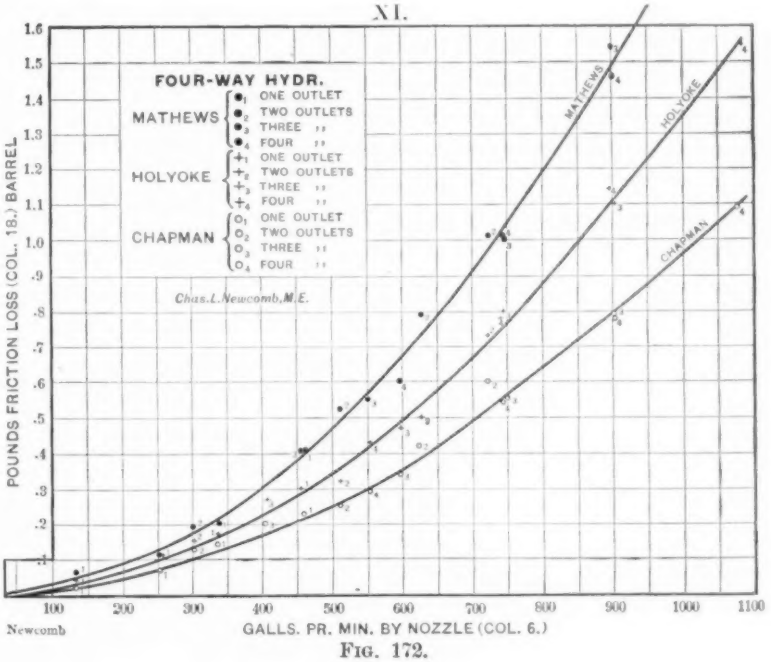


FIG. 172.

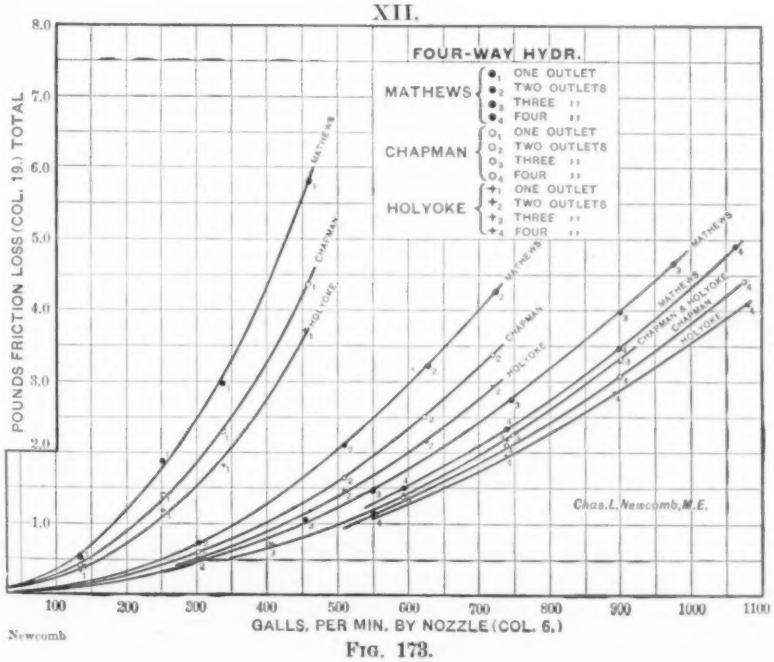


FIG. 173.

XIII

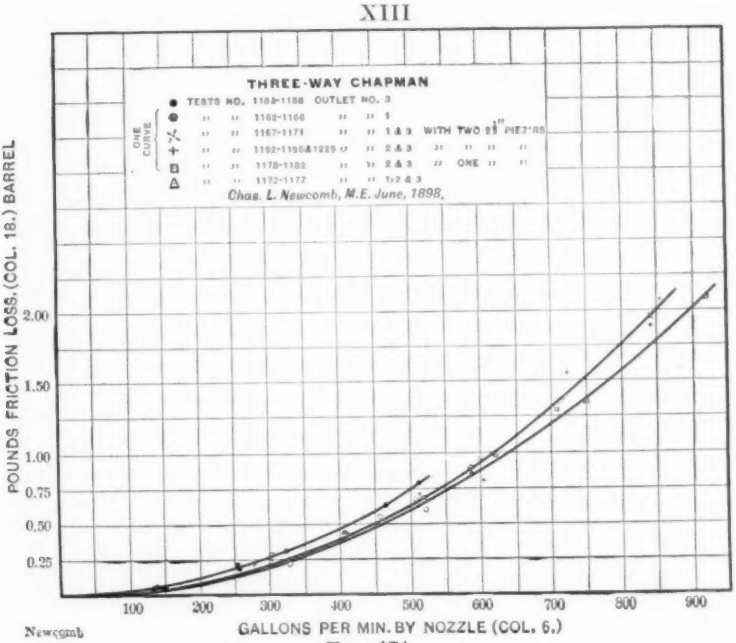


FIG. 174.

XIV

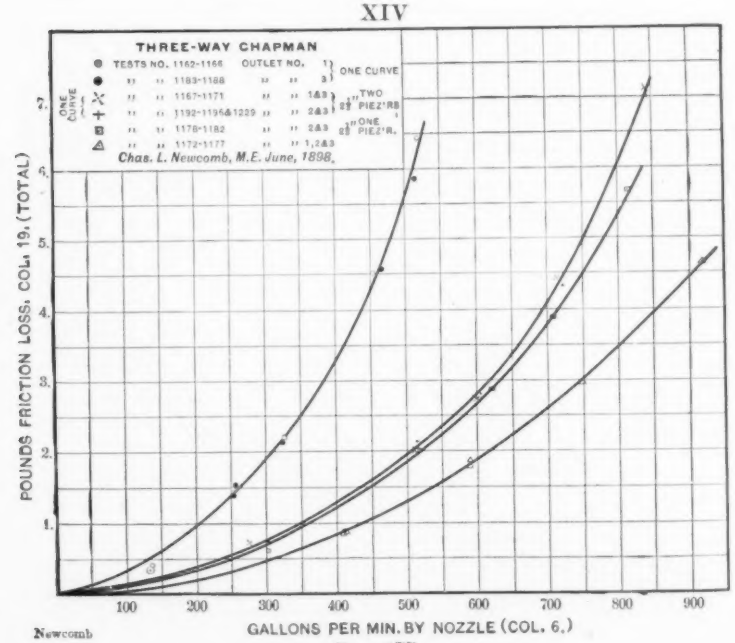


FIG. 175.

XV

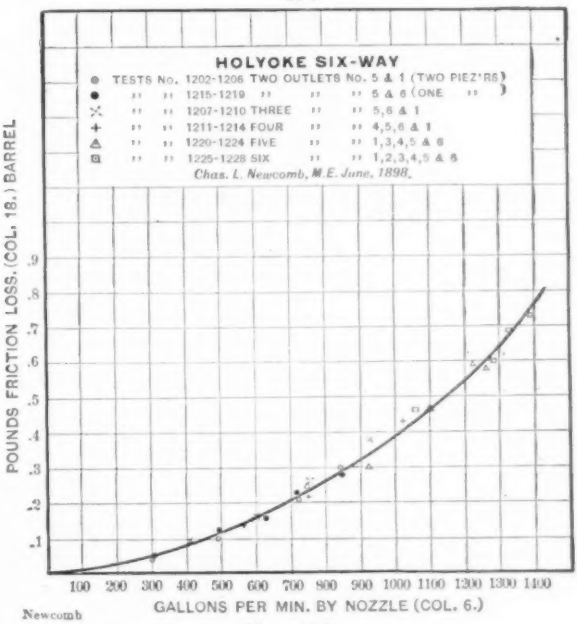


FIG. 176.

XVI

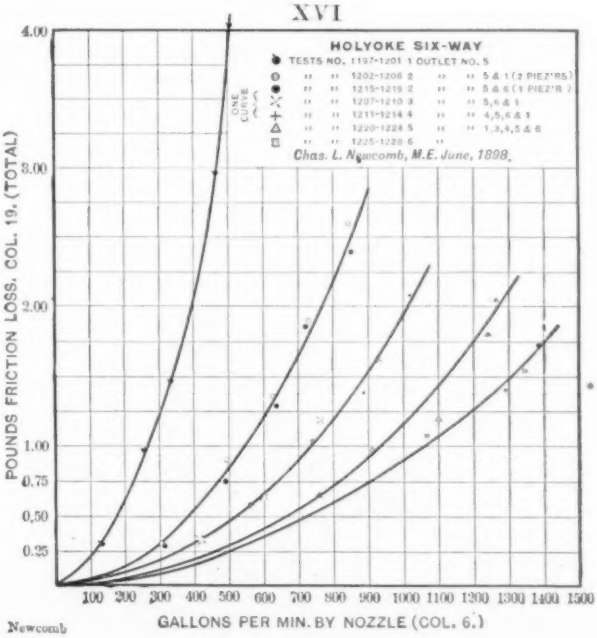


FIG. 177.

XVII

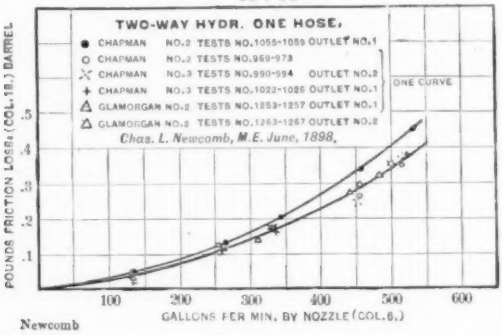


FIG. 178.

XVIII

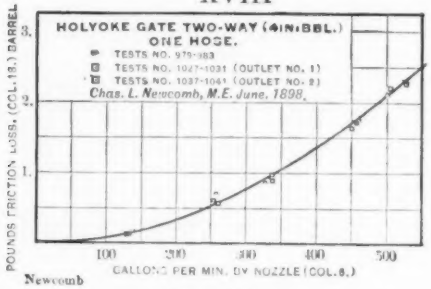


FIG. 179.

XIX

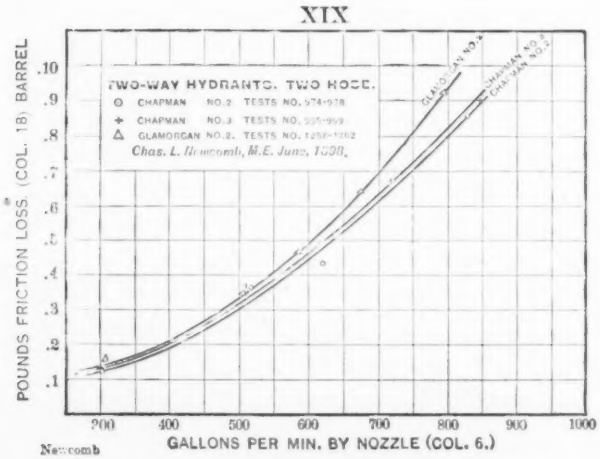


FIG. 180.

XX

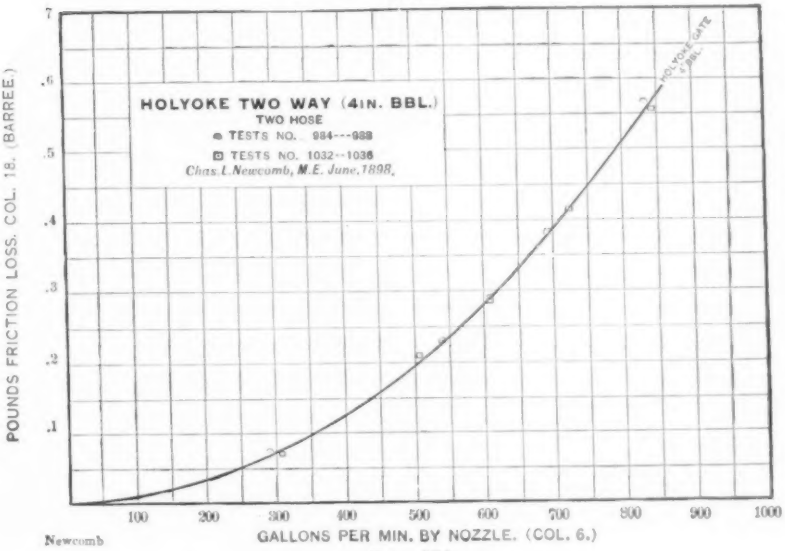


FIG. 181.

XXI

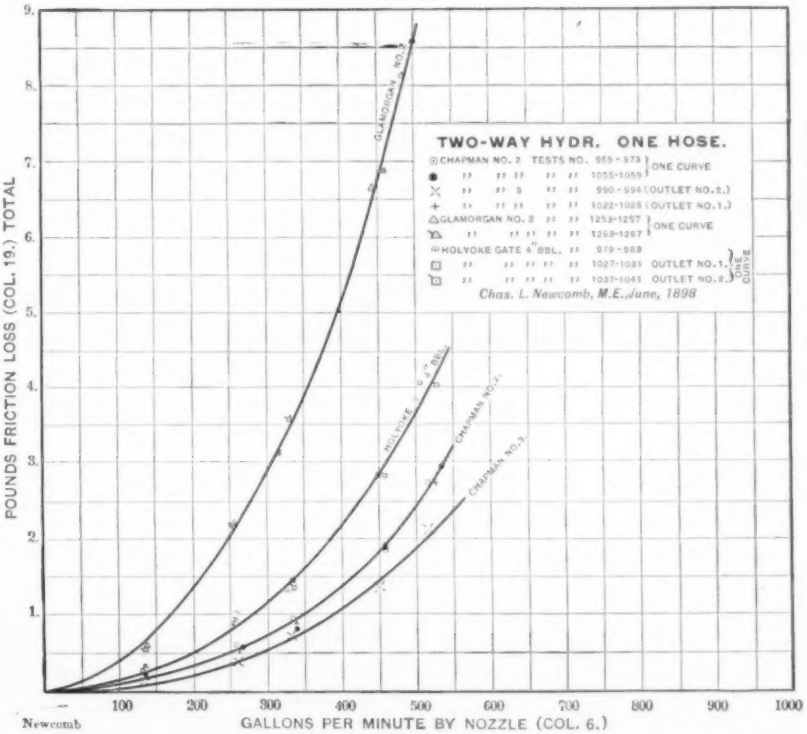


FIG. 182.

XXII

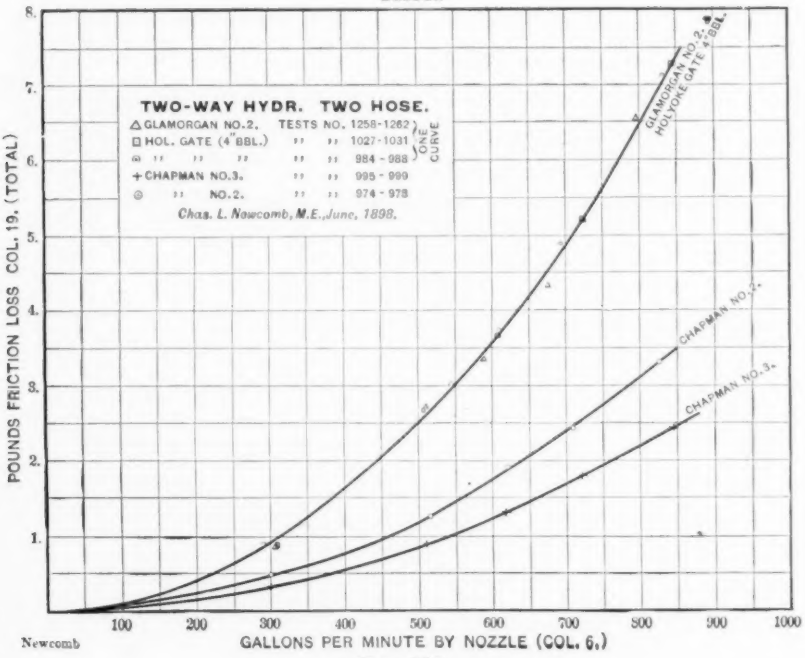

TWO-WAY HYDR. TWO HOSE.
△ GLAMORGAN NO.2. TESTS NO. 1258-1262
□ HOL. GATE (4" BBL.) " " 1027-1031
⊙ " " " " " 984 - 988
+ CHAPMAN NO.3. " " 995 - 999
⊙ " NO.2. " " 974 - 978
CONE CURVE
Chas. L. Newcomb, M.E., June, 1898.
GLAMORGAN NO.2.
HOLYOKE GATE 4" BBL.
CHAPMAN NO.2.
CHAPMAN NO.3.
POUNDS FRICTION LOSS COL. 19. (TOTAL)
GALLONS PER MINUTE BY NOZZLE (COL. 6.)
Newcomb

FIG. 183.

XXIII

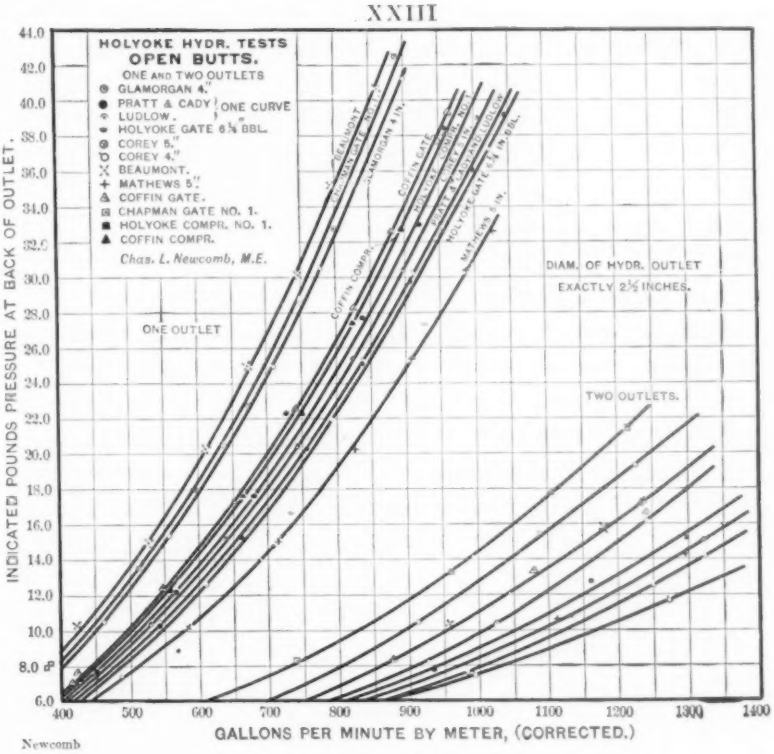


FIG. 184.

XXIV

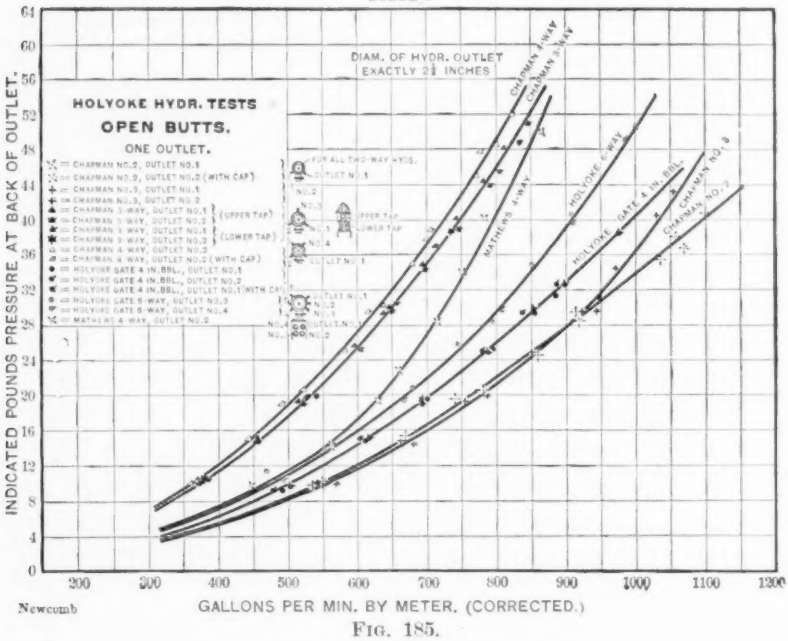


FIG. 185.

XXV

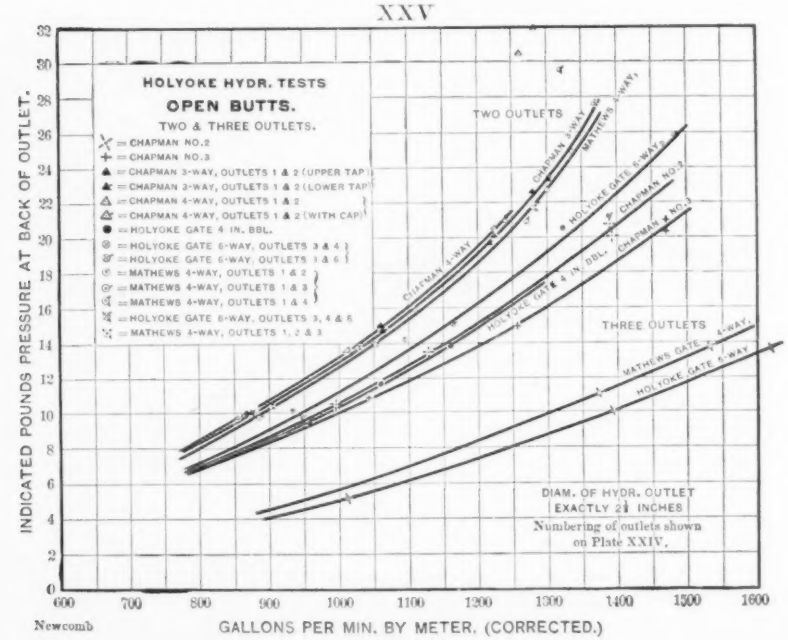


FIG. 186.

Plates XXIII. to XXV. (Figs. 184–5–6) give similar plottings for the open-butt tests, pressures at the back of the hydrant being taken for one coördinate and gallons per minute for the other.

These curves give convenient means of studying the general uniformity of results and for interpolating between the points given in the summary tables. They also furnish a complete graphical record of the results which are shown in tabulated form on the data sheets. As many curves as were consistent with clearness have been plotted on one sheet, and, in general, hydrants of similar main dimensions will be found plotted together.

Accuracy.—In the friction-loss tests the nozzle pressures were read to the nearest 0.1 of a pound, which represented a little less than one-quarter inch on the scale, and was therefore a very easily read division. The fluctuations under ordinary conditions, due to changes of pressure in the city mains, were less than one pound. When more than this, the unusual readings and the corresponding ones on the other gauges were thrown out. It may safely be assumed that the nozzle pressures were correct within one-quarter of a pound. At the pressures worked at, this would mean an error in quantity of less than 1 per cent.

The U-gauges in the later tests were read to the nearest 0.01 pound; in the earlier ones to the nearest 0.05 pound, except for the very small losses, where somewhat closer readings were made. At the small losses there was almost no vibration of the mercury columns, thus facilitating accurate readings. For the larger losses careful throttling of the cocks reduced the vibrations to a small range. By plotting these losses with the quantities corresponding, and locating an average curve through the points, the errors tend to neutralize each other, and it will be seen by studying the plates that the points do very readily locate such curves, and that but few of them lie far outside the average line. Losses read from these curves may be considered correct within 2 per cent. Table I. and the pyramid diagrams were made from the curves.

In the open-butt tests the mercury column was read to the nearest 0.1 pound. The average pressure from the ten readings would probably not be in error more than 0.2 pound and in general not more than 0.1 pound. In the tests where the mercury column was connected to the 6-inch piezometer the U-gauge error also enters, but in general this did not exceed about

0.1 pound. This means when using the U-tube a maximum error of 0.3 pound and without the tube 0.2 pound. The percentage errors decrease with the increase of pressures.

In estimating on the meter just when the dial hand, with the meter in motion, passes the zero point, an error of 1½ cubic feet might be made. If the errors at the beginning and end happen to be in the opposite direction, the maximum error would be 3 cubic feet. One hundred cubic feet was the smallest quantity passed in any test, and generally the quantity was considerably larger. Therefore in the worst case the percentage error in reading the meter does not exceed 3 per cent. The meters were calibrated by the nozzles, and an average curve plotted with meter readings for one coördinate, and nozzle quantities for the others. The errors in the meters were found to be practically constant for any given quantity, so that an error of over 1 per cent. in the calibration is improbable.

The stop-watch was read to the nearest one-fifth second, and was frequently rated so that with ordinary care the time should be correct within 1 per cent. and often closer. With the above errors happening to be all in one direction the determination of the discharge per minute in any one test might be 5 per cent. out. This would be the worst condition, and in general the error would be much less.

In this work, as in the friction-loss tests, the results were plotted and average curves drawn. The results from these curves may reasonably be expected to be correct within 3 per cent.

Considering all the above, it will be seen that the results are amply accurate for all practical purposes.

Hydrants Tested.

The following cuts, arranged in alphabetical order, show the general features and dimensions of the hydrants tested. A study of the results of the tests in connection with these cuts will show the reasons for the differences found.

BEAUMONT.—Fig. 187 shows the general appearance of the hydrant tested. The casting towards the outlets was well rounded and the nozzles leaded in, making a smooth joint.

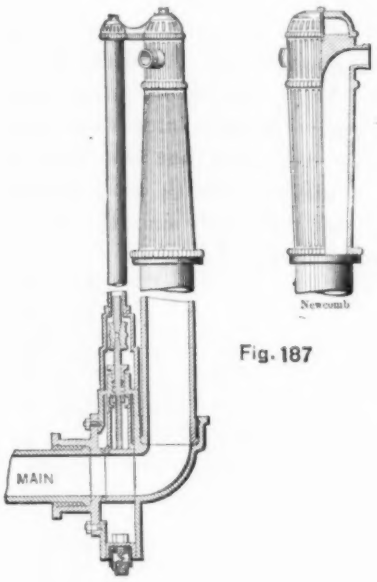


Fig. 187

Inside diameter of cylindrical barrel, 4⅛ inches.

Diameter of gate opening, 4⅛ inches.

Fig. 188 shows a cross-section of hydrant barrel at gauge connection.

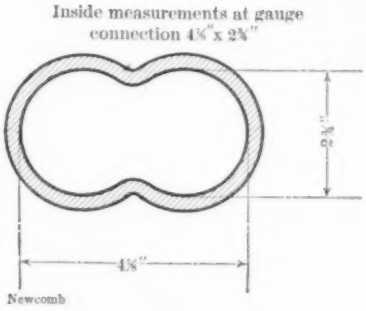


FIG. 188.

CHAPMAN NOS. 1 AND 2.—Fig. 189 shows the general appearance of hydrants Nos. 1 and 2, but the hydrants tested were simple two-ways with no steamer connections. The dimensions given are for the hydrants tested. No. 1 was the regular commercial hydrant, and had sharp, jagged corners at the 2½-inch outlets. In No. 2 the same casting was used, but the corners had been chipped and filed and made as smooth as possible, working from the outside. The result was fairly smooth curves, but of short radius, probably about ¼ inch.

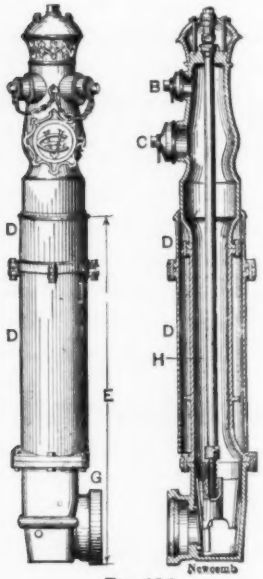


Fig. 189

Gauge connection: Inside diameter of barrel at point opposite *B*:

For No. 1 = 4⅝ inches
" No. 2 = 5 "

Smallest inside diameter of barrel, 5¼ inches.

Diameter of gate opening, 5 inches.

CHAPMAN No. 3.—Fig. 190 shows the new-pattern hydrant called No. 3. This was a two-way with steamer connection. The casting at 2½-inch outlets was smooth and well rounded. Nozzles did not butt close against casting, but left a groove-like space about $\frac{1}{16}$ inch across.

Fig. 190

Inside diameter of barrel at gauge connections, $7\frac{3}{4}$ inches.

Inside diameter of cylindrical barrel, 7 inches.

Gate opening, oval, $6\frac{3}{8}$ inches by $5\frac{7}{16}$ inches, equal in area to circle 5.9 inches diameter.

CHAPMAN THREE-WAY.—Fig. 191 shows the three-way independent gate hydrant. Nozzles projected into hydrant about $\frac{1}{4}$ inch, and had flat ends, making sharp corners. The independent gate arrangement differs somewhat from that in the four-way hydrant. The guides are cast with the head, and have rounded corners toward the current. The inside independent valve, when wide open, projects $\frac{1}{8}$ inch to $\frac{3}{16}$ inch into nozzle openings.

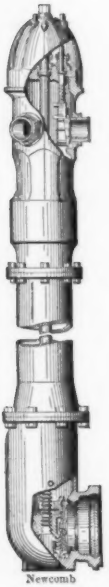


FIG. 191.

Barrel at gauge connection, hexagonal; distance inside between flat faces, $6\frac{9}{16}$ inches.

Round, inside diameter, 6 inches.

Cylindrical barrel, inside diameter, $6\frac{3}{8}$ inches.

Gate opening, oval, $5\frac{7}{16}$ inches by $4\frac{5}{16}$ inches.

CHAPMAN FOUR-WAY.—Fig. 192 shows the four-way independent gate hydrant. The lowest point on the independent gates projected about $\frac{1}{4}$ inch into openings when gates were wide open. This caused a noticeable breaking of the stream in the open-butt tests.

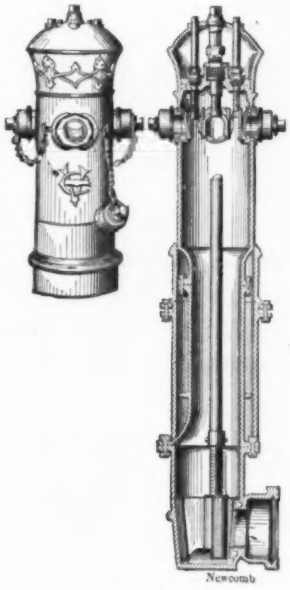


FIG. 192.

Gauge connection ; inside diameter of barrel, $8\frac{1}{4}$ inches.

Inside diameter of cylindrical barrel, $6\frac{3}{8}$ inches.

Diameter of gate opening, $6\frac{1}{16}$ inches.

COFFIN GATE.—Fig. 193 shows general appearance of Coffin hydrant, but the one tested had no steamer connection. Dimensions below are for hydrant tested. The nozzle entrances had well-rounded corners.

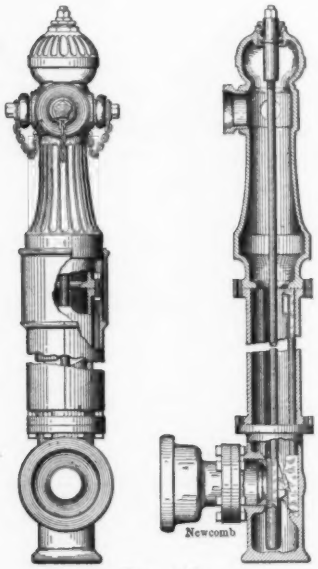


FIG. 193.

Inside diameter at gauge connection, 6 inches.

Diameter at contraction, $5\frac{3}{8}$ inches.

Inside diameter of cylindrical barrel, $6\frac{1}{2}$ inches.

Diameter of gate opening, $4\frac{3}{4}$ inches.

COFFIN COMPRESSION.—Fig. 194 shows the two-way hydrant with steamer connection tested. The 2½-inch nozzle entrances had well-rounded corners. This pattern has no contracted section like that in the gate hydrant.

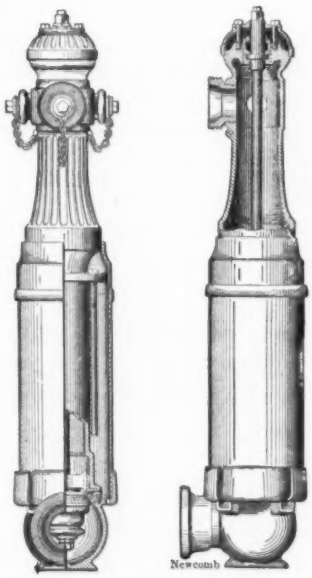


FIG. 194.

Inside diameter at gauge connection, 6 inches.

Inside diameter of cylindrical barrel, 6 inches.

Diameter of gate opening, 5 inches.

COREY 4-INCH AND 5-INCH.—Fig. 195 shows the general features of both 4-inch and 5-inch hydrants. They were both two-ways and identical, except for size. In this hydrant the barrel gauge connection was enough below the outlet to enter the cylindrical part of the barrel.

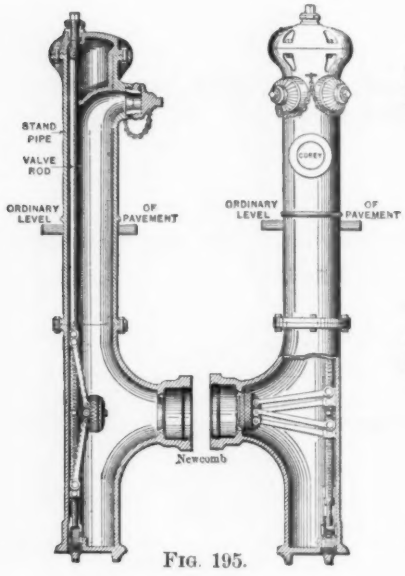


FIG. 195.

Inside diameters at gauge connection:
For 4-inch barrel = $5\frac{3}{4}$ inches
" 5-inch " = $6\frac{3}{4}$ "

Inside diameters of cylindrical barrel:
For 4-inch hydrant = $5\frac{3}{4}$ inches
" 5-inch " = $6\frac{3}{4}$ "

Diameters of gate opening:
For 4-inch hydrant = 4 inches
" 5-inch " = 5 "

GLAMORGAN.—Fig. 196 shows the general features of the Glamorgan hydrants referred to as Nos. 1 and 2. The castings of both hydrants were similar, except that the outlets of No. 1 were less than 180 degrees apart, or about like the ordinary hydrant. The nozzle outlets were not rounded, but presented square corners in the barrel.

No. 2 was fitted with independent cut-offs attached to outlets. These were of a peculiar design, working on the principle of a piston valve, the shutting off being accomplished by revolving an external collar which moved the valves over the ports. The ports compelled the water to make several turns over sharp corners, accounting for the large frictional loss found.

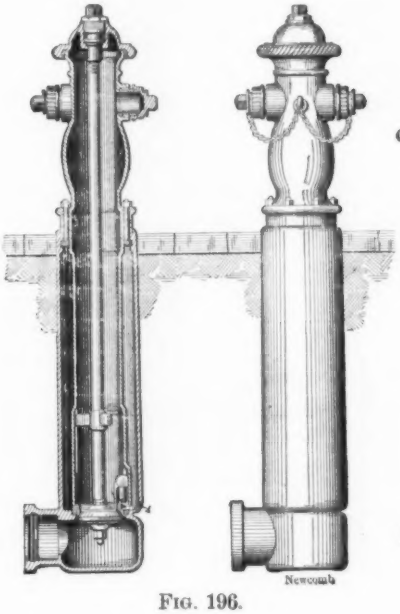


FIG. 196.

Inside diameters at gauge connection:
For No. 1, $4\frac{3}{8}$ inches
" No. 2, $6\frac{1}{2}$ "

Inside diameters of barrel at smallest part:
For No. 1, 5 inches
" No. 2, $6\frac{3}{4}$ "

Diameters of gate opening:
For No. 1, $4\frac{1}{4}$ inches
" No. 2, $6\frac{5}{8}$ "

HOLYOKE GATE (4-inch, $5\frac{1}{4}$-inch, and $6\frac{1}{4}$-inch Barrel).—Fig. 197 shows the general features of the nominal 4-inch, 5-inch, and 6-inch hydrants. These were all two-way. Casting at outlet had rounding corners, but was rough, with small nubbles and projections.

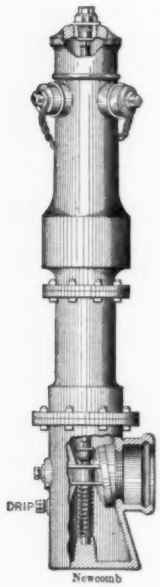


FIG. 197.

Inside diameters at gauge connections:
For 4-inch hydrant, $5\frac{9}{16}$ inches
" 5-inch " $6\frac{1}{8}$ "
" 6-inch " 7 "

Inside diameters of cylindrical barrel:
For 4-inch hydrant, $4\frac{3}{8}$ inches
" 5-inch " $5\frac{3}{8}$ "
" 6-inch " $6\frac{3}{8}$ "

Diameters of gate opening:
For 4-inch hydrant, $4\frac{1}{16}$ inches
" 5-inch " 5 "
" 6-inch " 6 "

HOLYOKE COMPRESSION NOS. 1 AND 2.—Fig. 198 shows the general features of hydrants Nos. 1 and 2, which were both two-way. The casting at outlet had rounding corners, but was rough, with small nubbles and projections.

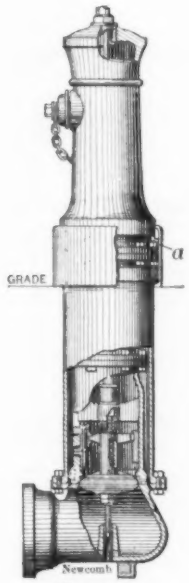


FIG. 198.

Inside diameters at gauge connections:
For No. 1, $6\frac{1}{4}$ inches
" No. 2, $5\frac{5}{8}$ "

Inside diameters of cylindrical barrel:
For No. 1, $5\frac{1}{4}$ inches
" No. 2, 5 "

Diameters of gate opening:
For No. 1, $5\frac{1}{2}$ inches
" No. 2, $4\frac{1}{2}$ "

HOLYOKE FOUR-WAY.—Fig. 199 shows the independent gate hydrant, the condition at the nozzles being as shown in the cut.

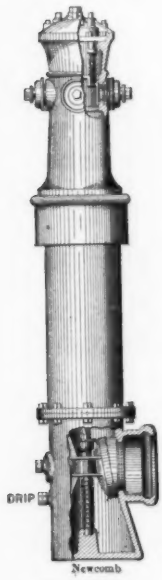


FIG. 199.

Inside diameter of barrel at gauge connection, $7\frac{1}{2}$ inches.

Inside diameter of cylindrical barrel, $6\frac{3}{8}$ inches.

Diameter of gate opening, 6 inches.

HOLYOKE SIX-WAY.—Fig. 200 shows the general appearance of the six-way independent gate hydrant tested. The main gate and barrel are of the same style as shown in Fig. 197. The independent gates at the nozzle outlets are similar to those shown in Fig. 199.

FIG. 200.

Inside diameters at gauge connections:
For upper connection, $9\frac{1}{8}$ inches
" lower " $9\frac{7}{8}$ "

Inside diameter of cylindrical barrel, 10 inches.

Diameter of gate opening, 8 inches.

LUDLOW.—Fig. 201 shows the two-way hydrant tested. The nozzle outlets were square-cornered, the sketch well showing the conditions.

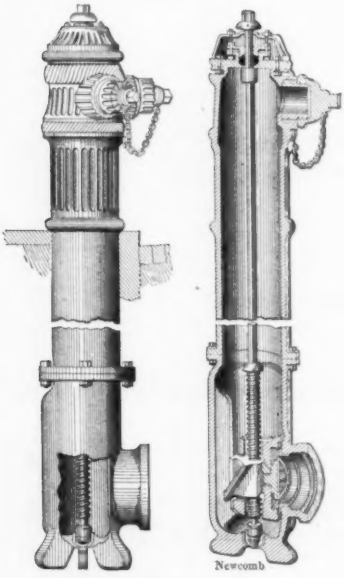


FIG. 201.

Inside diameter at gauge connection, 7⅝ inches.

Inside diameter of cylindrical barrel, 7 inches.

Diameter of gate opening, 5 inches.

MATHEWS 5-INCH.—Fig. 202 shows the general appearance of the two-way compression hydrant, with steamer connection, tested. The hydrant tested was fitted with the double-valve arrangement, as shown in Fig. 203. The casting of hydrant was rounded at the outlets to a radius of about $\frac{1}{4}$ inch, not having the square corners shown in the cut.

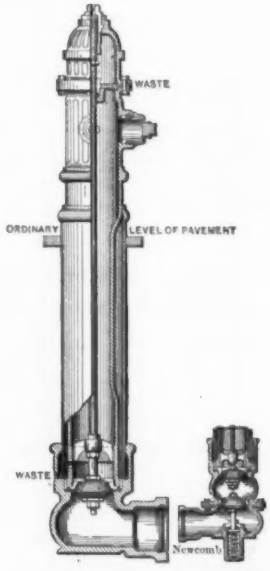


FIG. 202. FIG. 203.

Inside diameter at gauge connection, $7\frac{7}{16}$ inches.

Inside diameter of cylindrical barrel, $5\frac{3}{4}$ inches.

Diameter of both gate openings, $5\frac{5}{16}$ inches.

MATHEWS FOUR-WAY.—Fig. 204 shows the general appearance of a single-valve compression hydrant with independent gates at outlet. In Fig. 202 will be seen the main valve arrangement of the hydrant tested, and in Fig. 204 the design of the independent gates. In the hydrant tested the head was of somewhat different design to provide for the two additional outlets directly above those shown. With independent gates open the distance from valve face to edge of outlet is from $1\frac{1}{2}$ inches to $1\frac{5}{8}$ inches.

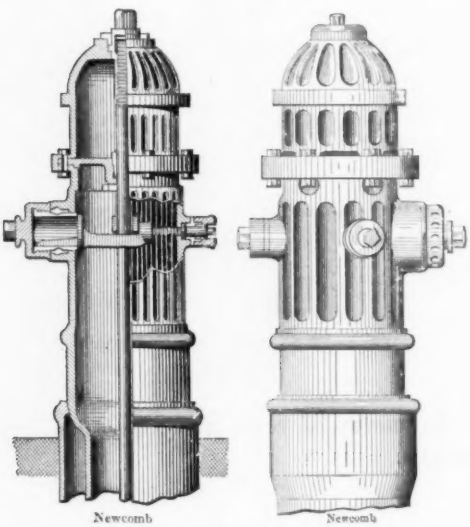


FIG. 204.

Inside diameter at gauge connection, $8\frac{1}{2}$ inches.

Inside diameter of cylindrical barrel, $6\frac{1}{4}$ inches.

Diameter of gate opening, $6\frac{3}{8}$ inches.

PRATT & CADY.—Fig. 205 shows the general appearance of the two-way hydrant, with steamer connection tested. The hydrant tested was fitted with independent gates for the 2½-inch outlets. These gates were moved up and down by spindles through the head, somewhat similar to the Chapman and Holyoke hydrants, but the gates themselves had rounding surfaces following the curvatures of the barrel. The casting at the outlets had square and rather ragged corners.

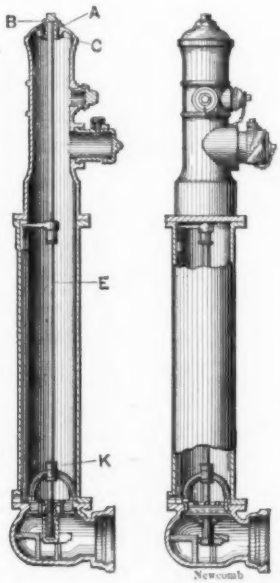


FIG. 205.

Inside diameter at gauge connections, 6 inches.

Diameter of cylindrical barrel, 8⅜ inches.

Diameter of gate inlet, 6¼ inches.

The Results in Brief.

To put the results into shape for ready use the tables following have been prepared: Tables I., II., and III. give the entire results of the friction-loss tests; Tables IV. and V. the discharging capacity of open butts for the range of pressures covered by the tests. These tables were filled out by readings directly from the curves, Plates I. to XXV., inclusive.

To further aid in making comparisons and to put the whole friction-loss data into shape to appeal quickly to the eye, the pyramid diagrams Figs. 206, 207, and 208 have been prepared. The points selected for these diagrams correspond to one or more standard fire streams; that is, 250 gallons per minute through each line of 2½-inch fire hose.

The open-butt tables are of value in testing water-works systems to determine their capacities at useful pressures where lack of either time or facilities prevents using more accurate apparatus. In such cases the open butt gives quickly and inexpensively fairly accurate results for this kind of work. Differences in design and construction of the outlet materially affect the discharge, but a study of the tables, together with the cuts of the hydrants, will enable one to apply intelligent corrections for outlets differing from any here tested.

In practice the gauge would often be connected into a tapped hole in a hydrant cap, and the cap screwed to one of the other outlets of the hydrant if it had more than one. This in all ordinary cases would give the same result as a connection tapped into the back of the hydrant, as was the case in most of the tests. The few tests in which the gauge was connected, first one way and then the other, show practically no difference in the result

TABLE I.

Hydrant Friction Losses—One Stream Flowing.

Name of Hydrant.	Friction Loss in Pounds with Various Rates of Flow. Gallons per Minute Flowing.														
	100			200			300			400			500		
	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.
	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
B'um't { L. H. outlet	0.07	0.25	**0.32**	0.23	0.76	**0.99**	0.49	1.49	**1.98**	0.85	2.68	**3.53**	1.36	4.89	**6.25**
B'um't { R. H. "	0.07	0.18	**0.25**	0.23	0.59	**0.82**	0.49	1.24	**1.73**	0.85	2.24	**3.09**	1.33	3.99	**5.35**
Chapman No. 1	0.01	0.19	**0.20**	0.07	0.69	**0.76**	0.17	1.47	**1.64**	0.32	2.74	**3.06**	0.51	4.98	**5.49**
" No. 2	0.03	0.05	**0.08**	0.08	0.26	**0.34**	0.16	0.58	**0.74**	0.27	1.11	**1.38**	0.40	2.05	**2.45**
" No. 3	0.02	0.01	**0.03**	0.07	0.15	**0.22**	0.14	0.41	**0.55**	0.22	0.87	**1.09**	0.34	1.55	**1.89**
Chapman 3-way	0.03	0.32	**0.35**	0.12	0.84	**0.96**	0.27	1.61	**1.88**	0.47	2.80	**3.27**	0.73	4.87	**5.60**
" 4-way	0.01	0.24	**0.25**	0.04	0.82	**0.86**	0.10	1.72	**1.82**	0.17	3.03	**3.20**	0.25	....	.....
Coffin Gate	0.06	0.08	**0.14**	0.22	0.31	**0.53**	0.48	0.72	**1.20**	0.86	1.37	**2.23**	1.44	2.48	**3.92**
" Compression	0.05	0.05	**0.10**	0.15	0.23	**0.38**	0.30	0.55	**0.85**	0.53	1.07	**1.60**	0.88	1.95	**2.83**
Corey 4-in.	0.07	0.04	**0.24**	0.23	0.19	**0.42**	0.49	0.44	**0.93**	0.85	0.93	**1.78**	1.37	1.76	**3.13**
" 5-"	0.01	0.03	**0.04**	0.07	0.18	**0.25**	0.17	0.41	**0.58**	0.32	0.73	**1.05**	0.51	1.21	**1.72**
Glamorgan, No. 1	0.05	0.10	**0.15**	0.11	0.39	**0.50**	0.19	0.87	**1.06**	0.31	1.63	**1.94**	0.49	2.83	**3.32**
" No. 2	0.02	0.37	**0.39**	0.07	1.29	**1.36**	0.14	2.73	**2.87**	0.22	4.88	**5.10**	0.34	8.28	**8.62**
Holyoke Gate, 4-in.	0.10	0.03	**0.13**	0.35	0.14	**0.49**	0.77	0.38	**1.15**	1.36	0.81	**2.17**	2.10	1.57	**3.67**
Holyoke Gate, 5½-in.	0.05	0.07	**0.12**	0.15	0.29	**0.44**	0.30	0.70	**1.00**	0.53	1.41	**1.94**	0.88	2.61	**3.49**
" " 6½-"	0.03	0.07	**0.10**	0.12	0.28	**0.40**	0.26	0.60	**0.86**	0.49	1.10	**1.59**	0.82	1.79	**2.61**
" Comp. No. 1	0.05	0.08	**0.13**	0.15	0.33	**0.48**	0.32	0.79	**1.11**	0.59	1.53	**2.12**	1.01	2.74	**3.75**
" " No. 2	0.07	0.24	**0.31**	0.26	0.33	**0.59**	0.56	0.72	**1.28**	1.03	1.32	**2.35**	1.68	2.42	**4.10**
Hol'ke Gate, 4-way	0.01	0.19	**0.20**	0.06	0.64	**0.70**	0.13	1.37	**1.50**	0.22	2.48	**2.70**	0.34	....	.....
" " 6-way	.....	....	**0.21**	.....	.....	**0.62**	.....	.....	**1.23**	.....	.....	**2.08**	.....	....	**3.85**
Ludlow	0.04	0.10	**0.14**	0.13	0.40	**0.53**	0.36	0.94	**1.20**	0.45	1.78	**2.23**	0.71	3.21	**3.92**
Mathews 5 in.	0.07	0.09	**0.16**	0.25	0.34	**0.59**	0.54	0.76	**1.30**	0.97	1.46	**2.43**	1.56	2.69	**4.25**
Mathews 4-way	0.03	0.29	**0.32**	0.09	1.09	**1.18**	0.18	2.22	**2.40**	0.31	3.99	**4.30**	0.47	....	.....
Pratt & Cady	0.02	0.12	**0.14**	0.07	0.39	**0.46**	0.14	0.87	**1.01**	0.23	1.68	**1.91**	0.34	2.92	**3.26**

TABLE II.

Hydrant Friction Losses—Two Streams Flowing.

Friction Loss in Pounds with Various Rates of Flow. Gallons per Minute Flowing.																		
300			400			500			600			700			800			
Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	
Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	
....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	
0.45	0.28	**0.73**	0.82	0.48	**1.30**	1.32	0.70	**2.02**	1.87	1.04	**2.91**	2.57	1.44	**4.01**	3.41	1.94	**5.35**	
0.08	0.74	**0.82**	0.17	1.32	**1.49**	0.30	1.97	**2.28**	0.44	2.79	**3.23**	0.61	3.86	**4.47**	0.78	5.24	**6 02**	
0.12	0.36	**0.48**	0.19	0.58	**0.77**	0.30	0.88	**1.18**	0.44	0.30	**1.74**	0.61	1.79	**2.40**	0.80	2.30	**3.10**	
0.13	0.19	**0.32**	0 20	0.33	**0.53**	0.31	0.51	**0.82**	0.46	0.75	**1.21**	0.63	1.04	**1.67**	0.82	1.36	**2.18**	
0.20	0.55	**0.75**	0.38	0.91	**1.29**	0.62	1.33	**1.95**	0.92	1.86	**2.78**	1.30	2.75	**4.05**	1.75	4.24	**5.99**	
0.10	0.48	**0 58**	0.17	0.84	**1.01**	0.25	1.33	**1.58**	0.36	1.95	**2.31**	0.49	2.69	**3.18**	0.64	....	.....	
0.50	0.25	**0.75**	0.82	0.54	**1.36**	1.26	0.86	**2.12**	1.82	1.20	**3.02**	2.48	1.61	**4.09**	3.23	2.25	**5.48**	
0.28	0.35	**0.63**	0.55	0.53	**1.08**	0.87	0.78	**1.65**	1.26	1.14	**2.40**	1.71	1.63	**3.34**	2.30	2.29	**4.59**	
0.42	0.23	**0.65**	0.75	0.32	**1.07**	1.16	0.47	**1.63**	1.62	0.73	**2.35**	2.18	1.06	**3.24**	2.78	1.49	**4.27**	
0.14	0.12	**0.26**	0.27	0.22	**0.49**	0.43	0.35	**0.78**	0.62	0.54	**1.16**	0.86	0.78	**1.64**	1.11	1.08	**2.19**	
0.17	0.42	**0.59**	0.26	0.67	**0.93**	0.37	1.07	**1.44**	0.51	1.60	**2.11**	0.69	2.26	**2.95**	0.92	3.01	**3.93**	
0.14	0.78	**0.92**	0.21	1 41	**1.62**	0.33	2.17	**2.50**	0.48	3.07	**3.55**	0.69	4.15	**4.84**	0.93	5.54	**6.47**	
0.72	0.20	**0.92**	1.24	0.38	**1.62**	1.94	0.56	**2.50**	2.81	0.74	**3.55**	3.87	0.97	**4.84**	5.10	1.37	**6.47**	
0.25	0.35	**0.60**	0.48	0.54	**1.02**	0.79	0.78	**1.57**	1.15	1.12	**2.27**	1.57	1.60	**3.17**	2.08	2.29	**4.37**	
0.25	0.17	**0.42**	0.47	0.30	**0.77**	0.74	0.48	**1.22**	1.00	0.68	**1.77**	1.48	1.00	**2.48**	.....	.....	**3.36**	
0.31	0.32	**0.63**	0.58	0.50	**1.08**	0.92	0.73	**1.65**	1.30	1.10	**2.40**	1.80	1.54	**3.34**	2.44	2.15	**4.59**	
....	....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	
0.13	0.35	**0.48**	0.22	0.65	**0.87**	0.34	1.03	**1.37**	0.49	1.51	**2.00**	0.67	2.08	**2.75**	0.88	...	.....	
0 05	0.26	**0.31**	0.08	0.47	**0.55**	0.12	0.73	**0.85**	0.16	1.05	**1.21**	0.21	1.45	**1.66**	0.26	1.97	**2.23**	
0.22	0.34	**0.56**	0.41	0.51	**0.92**	0.67	0.75	**1.42**	0.98	1.08	**2.06**	1.34	1.57	**2.91**	1.74	2.26	**4.00**	
0.57	0.18	**0.75**	0.95	0.41	**1.36**	1.47	0.65	**2.12**	2.12	0.90	**3.02**	2.82	1.17	**4.09**	3.83	1.65	**5.48**	
0.18	0.51	**0.69**	0.31	0.96	**1.27**	0.47	1.55	**2.02**	0.68	2.27	**2.95**	0.91	3.09	**4.00**	1.19	....	.....	
0.11	0.31	**0.42**	0.19	0.58	**0.77**	0.29	0.93	**1.22**	0.43	1.34	**1.77**	0.59	1.89	**2.48**	0.77	2.59	**3.36**	

TABLE III.

Hydrant Friction Losses—Three to Six Streams Flowing.

Name of Hydrant and Gallons per Minute Flowing.	No. of Streams Flowing.	Friction Loss in Pounds with Various Rates of Flow.														
		Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.	Barrel.	Nozzle.	Total.
Gallons per Min.		500			600			700			800			900		
Chapman 3-way	3	0.60	0.70	**1.30**	0.87	1.01	**1.88**	1.18	1.42	**2.60**	1.56	1.89	**3.45**	2.00	2.40	**4.40**
" 4-way	3	0.25	0.70	**0.95**	0.36	1.05	**1.41**	0.49	1.46	**1.95**	0.64	1.96	**2.60**	0.80	2.51	**3.31**
Holyoke 4-way	3	0.35	0.60	**0.95**	0.49	0.92	**1.41**	0.67	1.28	**1.95**	0.88	1.72	**2.60**	1.11	2.20	**3.31**
" 6-way	3	0.12	0.35	**0.47**	0.16	0.50	**0.66**	0.21	0.69	**0.90**	0.26	0.92	**1.18**	0.32	1.21	**1.53**
Mathews 4-way	3	0.47	0.73	**1.20**	0.68	1.07	**1.75**	0.91	1.49	**2.40**	1.19	1.97	**3.16**	1.49	2.51	**4.00**
Gallons per Min.		600			700			800			900			1,000		
Chapman 4-way	4	0.36	0.97	**1.33**	0.49	1.34	**1.83**	0.64	1.76	**2.40**	0.80	2.25	**3.05**	0.96	2.84	**3.80**
Holyoke 4-way	4	0.49	0.76	**1.25**	0.67	1.04	**1.71**	0.88	1.38	**2.26**	1.11	1.75	**2.86**	1.35	2.20	**3.55**
" 6-way	4	0.16	0.50	**0.66**	0.21	0.69	**0.90**	0.26	0.92	**1.18**	0.32	1.21	**1.53**	0.38	1.57	**1.95**
Mathews 4-way	4	0.68	0.84	**1.52**	0.91	1.16	**2.07**	1.19	1.53	**2.72**	1.49	2.04	**3.45**	1.81	....	**4 32**
Gallons per Min.		1,100			1,200			1,300			1,400			1,500		
Holyoke 6-way	5	0.46	0.98	**1.44**	0.54	1.21	**1.75**	0.63	1.47	**2.10**	0.75	....	....	0.91	...	....
" 6-way	6	0.46	0.61	**1.07**	0.54	0.71	**1.25**	0.63	0.85	**1.48**	0.75	1.00	**1.75**	0.91	1.14	**2.05**

TABLE IV.

FIG. 209.

DISCHARGE OF ONE OPEN HYDRANT BUTT.

These figures apply only approximately to hydrants in general, because slight variations in construction, even in hydrants from the same shop, considerably affect the discharge from the open butt. The cuts of hydrants will suggest corrections, so that with good judgment results accurate within from 10 to 20 per cent. may, in general, be obtained.

NAME OF HYDRANT.	DISCHARGE THROUGH ONE OPEN BUTT OF HYDRANT WITHOUT HOSE ATTACHED. (DIAMETER OF OUTLET EXACTLY 2½ INCHES.) GALLONS PER MINUTE. Hydrant Pressure indicated while Stream is Flowing, by Gauge attached to Hydrant as shown.* Pounds per Square Inch.								
	10	15	20	25	30	35	40	45	50
Beaumont	426	526	608	678	743	799	852	.....	....
Chapman No. 1	440	541	627	700	762	818	872	.....	....
" " 2	547	667	765	853	938	1,020	1,098	.....	....
" " 3	552	678	776	861	928	986	1,034	1,077	....
Chapman 3-way	371	458	531	593	648	701	748	792	834
" 4-way	363	444	513	573	630	680	725	767	806
Coffin Gate	497	607	701	780	850	913	974	.....	....
" Compression	501	612	709	790	860	924	985	.....	....
Corey 5-in	520	640	737	820	893	962	1,025	.....	....
Glamorgan 4-in	449	550	636	709	773	831	886	.....	....
Holyoke Gate, 4-in. bbl	500	612	703	783	857	927	994	1,058	....
" " 6¼-in. bbl.	545	663	760	844	921	992	1,048	.....	....
Holyoke Compr'n No. 1	512	628	723	804	876	942	1,003	.....	....
" Gate, 6-way	473	579	668	743	805	860	908	954	997
Ludlow	533	653	752	836	912	982	1,049	.....	....
Mathews 5-in	576	712	815	903	984	.....	.....	.....	....
Mathews 4-way	465	565	633	683	724	762	796	827	857
Pratt & Cady	533	653	752	836	912	982	1,049	.....	....

* Pressure can be equally well measured by tap at back of barrel opposite outlets.

TABLE V.

Newcomb

Discharge of Two and Three Open Hydrant Butts.

These figures apply only approximately to hydrants in general, because slight variations in construction, even in hydrants from the same shop, considerably affect the discharge from the open butt. The cuts of hydrants will suggest corrections, so that with good judgment results accurate within from 10 to 20 per cent. may, in general, be obtained.

Name of Hydrant.	No. of Outlets Open.	Discharge through Open Butts of Hydrant without Hose Attached. (Diameter of Outlet Exactly 2½ Inches.) Gallons per Minute. Hydrant Pressure indicated while Stream is Flowing, by Gauge attached to Hydrant Barrel as shown.* Pounds per Square Inch.				
		5	10	15	20	25
Beaumont	2	.....	904	1,158	1,327	.....
Chapman No. 1	2	.....	827	1,020	1,173	.....
" " 2	2	.....	974	1,197	1,377	.....
" " 3	2	.....	997	1,252	1,448	.....
Chapman 3-way	2	.....	876	1,074	1,223	1,329
" 4-way	2	.....	868	1,063	1,212	.....
Coffin Gate	2	.....	904	1,158	1,327	.....
Corey 4-in	2	.....	1,170	.....	.....	.....
Corey 5-in	2	.....	1,091	1,322	.....	.....
Glamorgan 4-in	2	.....	894	1,082	1,248	.....
Holyoke Gate, 4-in. bbl	2	.....	985	1,202	.....	.....
" " 6¼-in. bbl	2	.....	1,125	1,368	.....	.....
Holyoke Compression No. 1	2	.....	1,051	1,278	.....	.....
" Gate, 6-way	2	.....	942	1,152	1,326	1,466
Ludlow	2	.....	1,008	1,200	1,364	.....
Mathews 5-in	2	.....	1,091	1,322	.....	.....
Mathews 4-way	2	.....	890	1,089	1,238	1,341
Holyoke Gate, 6-way	3	992	1,388	.....	.....	.....
Mathews 4-way	3	940	1,304	1,596	.....	.....

* Pressure can be equally well measured by tap at back of barrel opposite outlets.

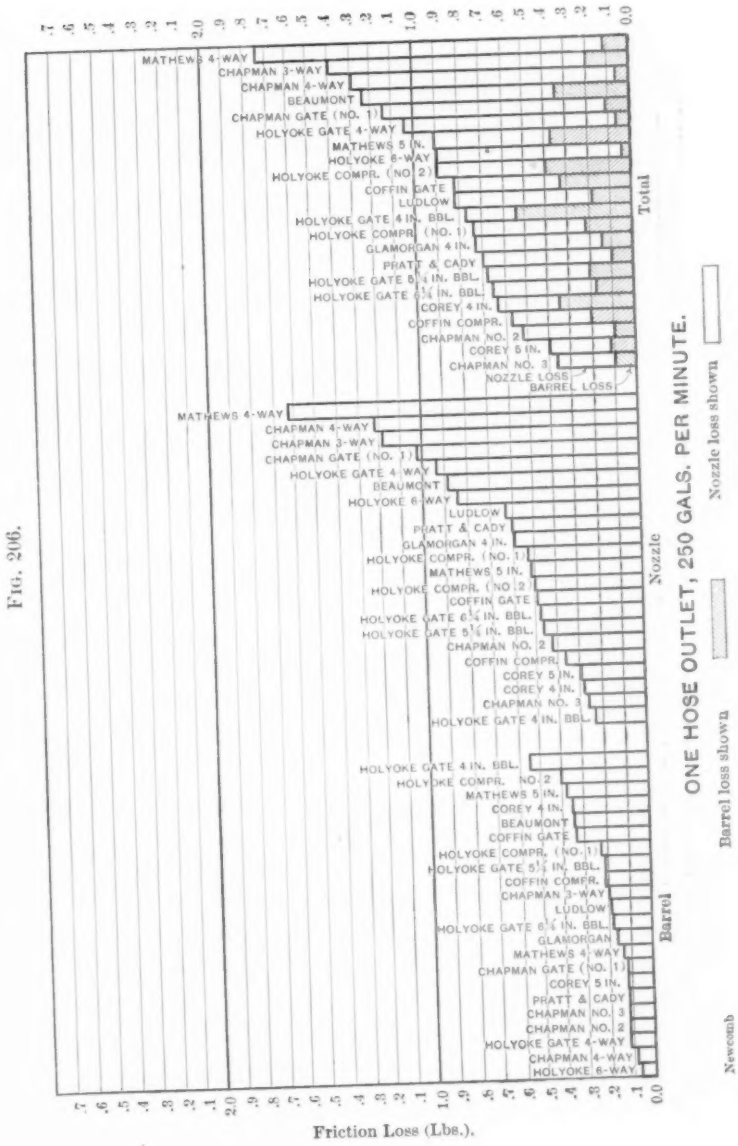

ONE HOSE OUTLET, 250 GALS. PER MINUTE.
Nozzle loss shown
Barrel loss shown
Total
Nozzle
Barrel
Newcomb
MATHEWS 4-WAY
CHAPMAN 3-WAY
CHAPMAN 4-WAY
BEAUMONT
CHAPMAN GATE (NO. 1)
HOLYOKE GATE 4-WAY
MATHEWS 5 IN.
HOLYOKE 6-WAY
HOLYOKE COMPR. (NO. 2)
COFFIN GATE
LUDLOW
HOLYOKE GATE 4 IN. BBL.
HOLYOKE COMPR. (NO. 1)
GLAMORGAN 4 IN.
PRATT & CADY
HOLYOKE GATE 6¼ IN. BBL.
HOLYOKE GATE 5¼ IN. BBL.
COREY 4 IN.
COFFIN COMPR.
CHAPMAN NO. 2
COREY 5 IN.
CHAPMAN NO. 3
NOZZLE LOSS
BARREL LOSS
MATHEWS 4-WAY
CHAPMAN 4-WAY
CHAPMAN 3-WAY
CHAPMAN GATE (NO. 1)
HOLYOKE GATE 4-WAY
BEAUMONT
HOLYOKE 6-WAY
LUDLOW
PRATT & CADY
GLAMORGAN 4 IN.
HOLYOKE COMPR. (NO. 1)
MATHEWS 5 IN.
HOLYOKE COMPR. (NO. 2)
COFFIN GATE
HOLYOKE GATE 6¼ IN. BBL.
HOLYOKE GATE 5¼ IN. BBL.
CHAPMAN NO. 2
COFFIN COMPR.
COREY 5 IN.
COREY 4 IN.
CHAPMAN NO. 3
HOLYOKE GATE 4 IN. BBL.
HOLYOKE GATE 4 IN. BBL.
HOLYOKE COMPR. NO. 2
MATHEWS 5 IN.
COREY 4 IN.
BEAUMONT
COFFIN GATE
HOLYOKE COMPR. (NO. 1)
HOLYOKE GATE 5¼ IN. BBL.
COFFIN COMPR.
CHAPMAN 3-WAY
LUDLOW
HOLYOKE GATE 6¼ IN. BBL.
GLAMORGAN
MATHEWS 4-WAY
CHAPMAN GATE (NO. 1)
COREY 5 IN.
PRATT & CADY
CHAPMAN NO. 3
CHAPMAN NO. 2
HOLYOKE GATE 4-WAY
CHAPMAN 4-WAY
HOLYOKE 6-WAY
2.0 1.9 1.8 1.7 1.6 1.5 1.4 1.3 1.2 1.1 1.0 .9 .8 .7 .6 .5 .4 .3 .2 .1 0.0
Friction Loss (Lbs.).

Fig. 206.

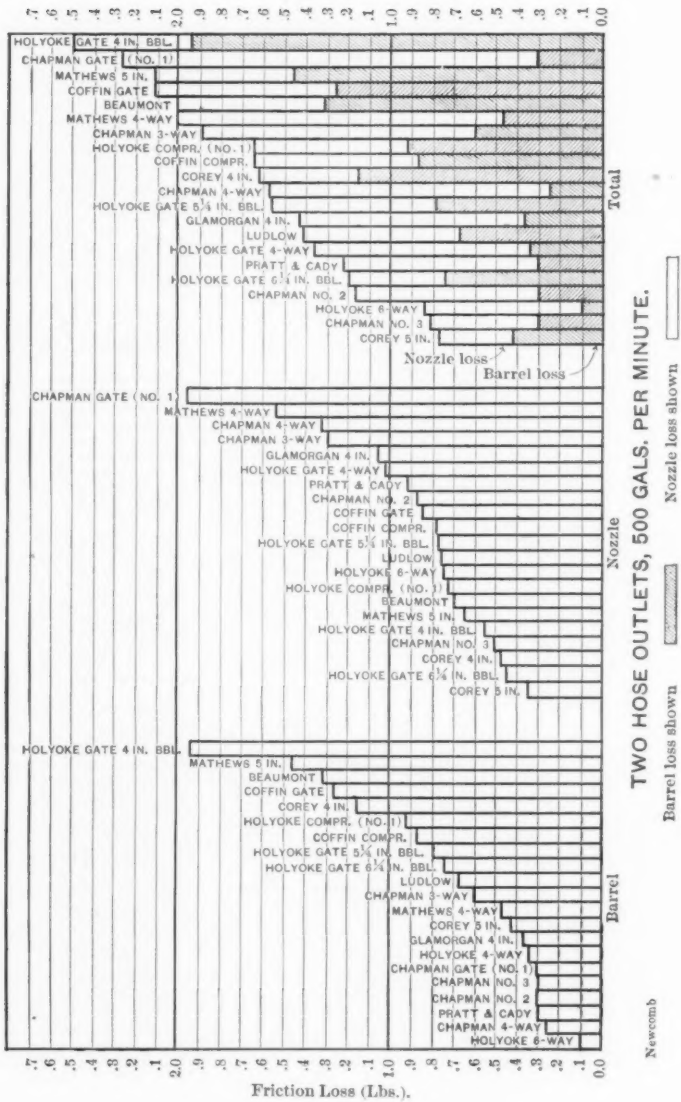

.7 .6 .5 .4 .3 .2 .1 2.0 .9 .8 .7 .6 .5 .4 .3 .2 .1 1.0 .9 .8 .7 .6 .5 .4 .3 .2 .1 0.0
HOLYOKE GATE 4 IN. BBL.
CHAPMAN GATE (NO. 1)
MATHEWS 5 IN.
COFFIN GATE
BEAUMONT
MATHEWS 4-WAY
CHAPMAN 3-WAY
HOLYOKE COMPR. (NO. 1)
COFFIN COMPR.
COREY 4 IN.
CHAPMAN 4-WAY
HOLYOKE GATE 5¼ IN. BBL.
GLAMORGAN 4 IN.
LUDLOW
HOLYOKE GATE 4-WAY
PRATT & CADY
HOLYOKE GATE 6¼ IN. BBL.
CHAPMAN NO. 2
HOLYOKE 6-WAY
CHAPMAN NO. 3
COREY 5 IN.
Total
Nozzle loss
Barrel loss
CHAPMAN GATE (NO. 1)
MATHEWS 4-WAY
CHAPMAN 4-WAY
CHAPMAN 3-WAY
GLAMORGAN 4 IN.
HOLYOKE GATE 4-WAY
PRATT & CADY
CHAPMAN NO. 2
COFFIN GATE
COFFIN COMPR.
HOLYOKE GATE 5¼ IN. BBL.
LUDLOW
HOLYOKE 6-WAY
HOLYOKE COMPR. (NO. 1)
BEAUMONT
MATHEWS 5 IN.
HOLYOKE GATE 4 IN. BBL.
CHAPMAN NO. 3
COREY 4 IN.
HOLYOKE GATE 6¼ IN. BBL.
COREY 5 IN.
Nozzle
HOLYOKE GATE 4 IN. BBL.
MATHEWS 5 IN.
BEAUMONT
COFFIN GATE
COREY 4 IN.
HOLYOKE COMPR. (NO. 1)
COFFIN COMPR.
HOLYOKE GATE 5¼ IN. BBL.
HOLYOKE GATE 6¼ IN. BBL.
LUDLOW
CHAPMAN 3-WAY
MATHEWS 4-WAY
COREY 5 IN.
GLAMORGAN 4 IN.
HOLYOKE 4-WAY
CHAPMAN GATE (NO. 1)
CHAPMAN NO. 3
CHAPMAN NO. 2
PRATT & CADY
CHAPMAN 4-WAY
HOLYOKE 6-WAY
Barrel
.7 .6 .5 .4 .3 .2 .1 2.0 .9 .8 .7 .6 .5 .4 .3 .2 .1 1.0 .9 .8 .7 .6 .5 .4 .3 .2 .1 0.0
Friction Loss (Lbs.).
TWO HOSE OUTLETS, 500 GALS. PER MINUTE.
Barrel loss shown
Nozzle loss shown
Newcomb

Fig. 207.

MATHEWS 4-WAY
CHAPMAN 4-WAY
HOLYOKE 4-WAY
HOLYOKE 6-WAY
Total
CHAPMAN 4-WAY
MATHEWS 4-WAY
HOLYOKE 4-WAY
HOLYOKE 6-WAY
Nozzle
MATHEWS 4-WAY
HOLYOKE 4-WAY
CHAPMAN 4-WAY
HOLYOKE 6-WAY
Barrel
FOUR HOSE OUTLETS, 1000 GALS. PER MIN.
Nozzle loss shown
Barrel loss
Nozzle loss
CHAPMAN 3-WAY
MATHEWS 4-WAY
HOLYOKE 4-WAY
CHAPMAN 4-WAY
HOLYOKE 6-WAY
Total
MATHEWS 4-WAY
CHAPMAN 4-WAY
CHAPMAN 3-WAY
HOLYOKE 4-WAY
HOLYOKE 6-WAY
Nozzle
CHAPMAN 3-WAY
MATHEWS 4-WAY
HOLYOKE 4-WAY
CHAPMAN 4-WAY
HOLYOKE 6-WAY
Barrel
THREE HOSE OUTLETS, 750 GALS. PER MIN.
In Totals, Barrel loss shown
Newcomb
5.0 .8 .6 .4 .2 4.0 .8 .6 .4 .2 3.0 .8 .6 .4 .2 2.0 .8 .6 .4 .2 1.0 .8 .6 .4 .2 0.0
Friction Loss (Lbs.).

FIG. 208.

DISCUSSION OF RESULTS.

The data and results being presented in full give ample chances for complete studies. Under these conditions detailed comparisons of the different hydrants have not been considered necessary or wholly desirable. A few general features may, however, be considered to advantage.

Barrel Loss.—The best point of comparison for the two-way hydrants is with two hose outlets in use and 500 gallons per minute flowing, as this represents the full normal capacity of the hydrant. Similar full-capacity points should be taken in comparing the larger hydrants.

To make clear the composition of the so-called barrel loss, the following table has been made and shows about how much of this loss occurs in the barrels proper. The figures show at once that a large part of the loss must be in the gate, and the sharp turn just beyond it, thus suggesting where to look for explanation for part of the large difference found. The distance from the centre of the main gate to the centre of the nozzles is on the average hydrant about 6½ feet. The following table gives approximately the friction loss in 6½ feet of clean, straight cylindrical pipe of about the same smoothness as the inside of the average hydrant barrel.

Nominal Diameter of Pipe (Inches).	Friction Loss in 6½ Feet. Pounds per Square Inch.			
	Gallons per Minute Flowing.			
	250	500	750	1,000
4	0.11	0.41	0.62	1.54
5	0.037	0.13	0.22	0.50
6	0.012	0.059	0.098	0.21
8	0.004	0.016	0.034	0.058

This table was made up from tests on ordinary new clean wrought-iron pipe, with 25 per cent. added to the wrought-iron pipe figures for the somewhat greater roughness of the inside of the average hydrant casting. The 25 per cent. was an assumption based on general experience with pipes of various degrees of roughness.

Comparing on the pyramid diagrams hydrants having two

2½-inch outlets, the difference in barrel losses is seen to be large. It is at once apparent that the 4-inch gate* and barrel are too small for a two-stream hydrant. A 4-inch barrel and a discharge of 500 gallons per minute mean a velocity of about 13 feet per second, so that a short length of barrel with the smoothest sort of a turn at the bottom develops an unreasonable loss, and this loss becomes still larger with the ordinary gate arrangement and sharp-turn bend.

Comparing further the hydrants which have about the same general dimensions, the difference in loss is considerable. This must be accounted for largely by differences in the design of the gate and the water passages in the immediate vicinity. Sharp corners, restricted sections, and sudden changes in the area of the passages all tend to produce eddyings, which use up pressure.

Nozzle Loss.—As already stated, the nozzle losses are pure friction losses, full correction for velocity having been made. To compare the outlets themselves, looking at them as simple nozzles, the condition with one stream on and 250 gallons per minute flowing is the best point. Studying the losses and the cuts together, the effect of sharp, jagged corners at the outlets is immediately seen, and the very material reduction in loss by even a slight rounding of the outlet is apparent.

Considering the hydrant as a whole, by taking the conditions with all of the outlets in use, it is seen that the average nozzle loss when all are in use is generally greater than with a single outlet in use. Separate tests on the two outlets of several hydrants showed a difference in loss with the same quantity flowing. This would account for a small part of the difference between loss with one outlet in use and with all outlets in use. Most of the difference is, however, undoubtedly due to the increase in choking and eddying effects at the top of the hydrant, with the higher velocities. A part of this is due to reactions from the eddies at the outlets and a part to the construction of the hydrant head.

The somewhat high losses with the Beaumont hydrant, considering the rounding outlets, is probably due to the fact that

*It was desired to have tests on some 4-inch-barrel hydrants to make the data complete, though most of the manufacturers would in general furnish larger barrels for two outlets. The 4-inch hydrant is, however, occasionally used.

the gauge connection was necessarily several inches below the outlet, so that there was considerable length between the gauge connection and the outlet, in which length some ordinary friction loss would occur.

Total Losses.—The total-loss pyramid shows the relative obstruction caused by the different hydrants taken as a whole. The part of this due to the barrel has been cross-hatched and the part due to the nozzle left white, thus showing at a glance the relation of the two factors making up the total loss.

It is desirable that the waste of pressure through a hydrant should be as small as it is practicable to make it. In high-pressure systems the losses found for the average hydrants are perhaps tolerable, but in lower-pressure systems—and many systems having a nominally high pressure become low under heavy draughts—every avoidable loss is objectionable. In the hydrants without independent gates, a simple rounding of the corners of the core at the outlet will make a material improvement in the nozzle losses. Reduction of loss for independent gate hydrants is more difficult, but some improvement is probably possible without serious trouble. The fact that some makers have found out how to reduce the barrel losses, so called, to comparatively small amounts is good working ground for improvements in those hydrants now having rather large losses.

It is not to be understood that this friction loss is the criterion for a perfect hydrant. Certainty of action under all conditions is of the greatest importance, but, other things being equal, the hydrant having the smallest total friction loss when working at its full capacity is the best.

WATER-HAMMER TESTS.

To get some measure of the water-hammer effect, caused when a hydrant is quickly shut down, the following apparatus was devised :

An indicator was attached to the connection in the 6-inch hydrant inlet which had been used by the U-tubes. The drum of this indicator was operated by clockwork in the manner shown in Fig. 210, and was so adjusted as to revolve once in from $3\frac{1}{2}$ to 4 minutes. A small weight and cord kept the pencil in contact with the paper during the taking of a card. Fig. 159 also shows the rig on a smaller scale.

Two lines of hose were taken from the hydrant under test to

the meter nozzle, and the hydrant gate opened wide. A $1\frac{7}{8}$-inch nozzle was used, and the pressure set at about 43 pounds by

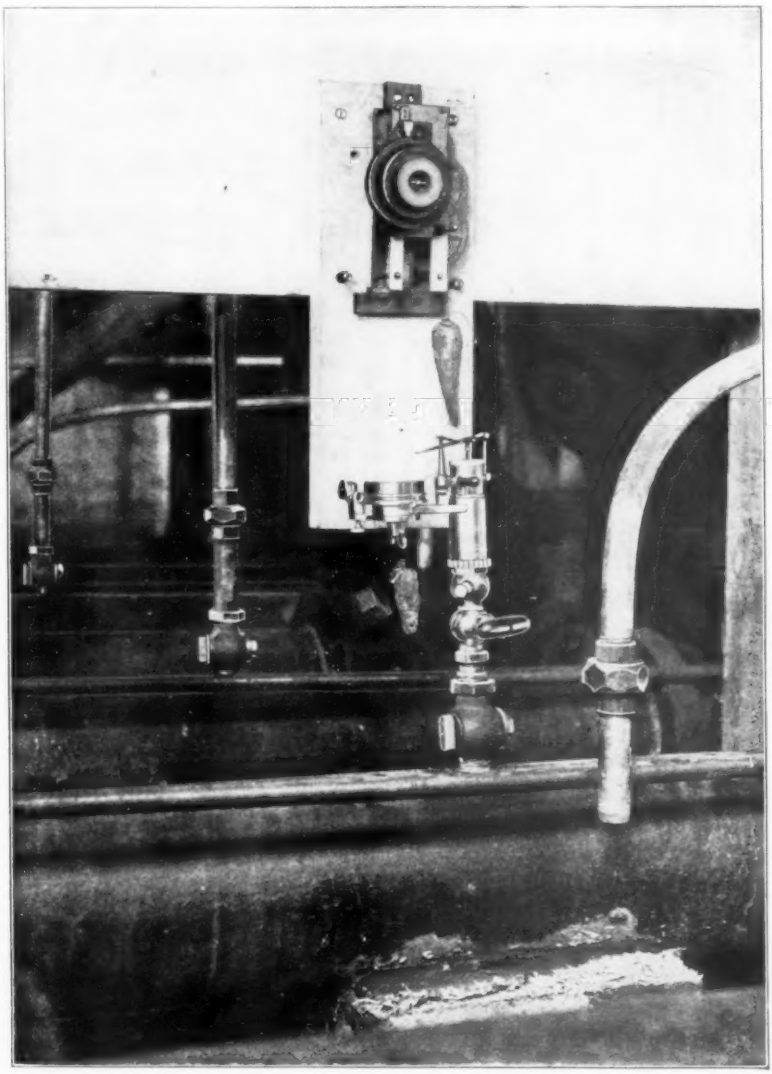
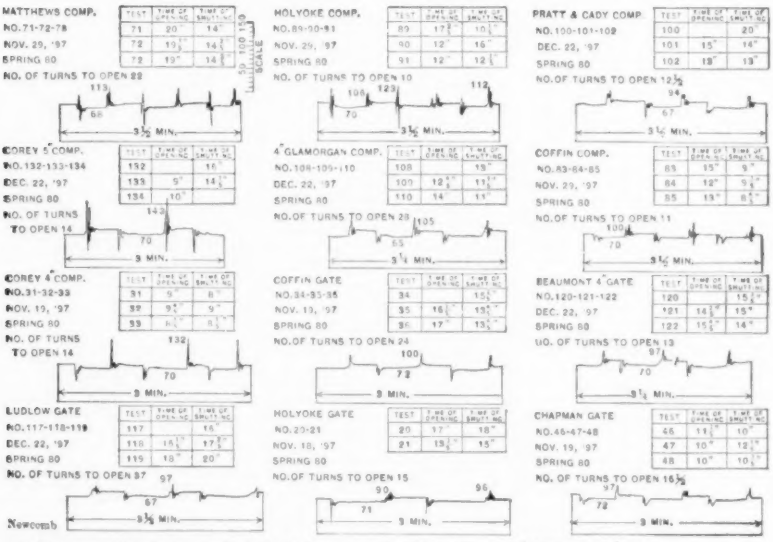


Fig. 210.—Apparatus for Taking Water-hammer Cards.

adjusting the gate on the inlet pipe at the entrance to the building. This gave a discharge of approximately 600 gallons per minute, which was considered a fair rate of flow for these tests.

At a given signal the indicator cock was opened, the clockwork set in motion, and time taken. After running steadily for about one-fourth of a minute, the hydrant valve was quickly closed; that is, it was closed as quickly as an active man could close it, using an ordinary hydrant wrench.

As soon as steady conditions were restored, the valve was as quickly opened again. This was repeated two to three times during the time that drum of indicator was in motion. Any variation of pressure caused by this closing or opening of the valve was recorded on the card. The time occupied in closing, and also



* Gate went hard near end of closing, preventing normal water-hammer action.

FIG. 211.—WATER-HAMMER CARDS.

in opening, was taken by a stop-watch, so that some comparison can be made between the treatment given any one hydrant and another. Sample cards from the various tests have been picked out, grouped, and directly reproduced in Fig. 211. In getting this reproduction the delicate lines made by the pencil point of the indicator were carefully gone over with ink so as to get a card which could be photographed.

This method of measuring cannot show just what the actual pounds rise of pressure in any particular case would be, as this depends on the arrangement of piping supplying the hydrant, and the drafts of water in the system independent of the hy-

drant draft. It does, however, give a rough means of comparing the different hydrants.

The cards show that the gate hydrants give generally considerably less water-hammer than the compression type.*

It was stated in explanation of the large hammer effects found in one of the special type of compression hydrants that firemen demand a hydrant which will give the full pressure at the earliest possible moment, so that a quick movement was purposely designed. It is firmly believed that in the long run it will be very much better to insure the safety of water mains by avoiding heavy water-hammer effects than to gain a few seconds of time at the risk of crippling the distributing system, very likely at the critical point of a fire.

In several of the compression hydrants a vibrating effect was noticed during the opening of the valve. This was apparently due to a chattering of the valve, and produced a rather undesirable water-hammer, as it had the effect of a series of quick, sharp blows.

CONSTRUCTION, STRENGTH, ETC.

Time prevented carrying on this part of the work as thoroughly as was desired, so that only partial results were obtained for some of the hydrants, while on most of them a considerably further study of construction and general features would have been desirable. What data were obtained is shown in Table VI., about which the following explanations may be made:

Column 3.—The inside volume of the hydrant is the volume between the main gate and the caps on the nozzles. This was obtained by filling the hydrant completely full of water and computing the volume from the weight of water.

Column 4.—The time to drain the hydrant was taken from the instant the main gate was closed. Facts which developed as the tests progressed from the ordinary handling of the hydrants tended to show the desirability of having positive-motion drip-

* The comparatively small hammer shown by the Coffin Compression is probably explained by the special double-beat valve, which is designed to give a more gradual closing than is ordinarily obtained with the compression type of gate.

The small hammer shown by the Pratt & Cady Compression is probably largely due to the fact that, when near the point of closing, the stem began to go hard, preventing a quick shutdown. The results are, therefore, not fairly comparable with the other hydrants. In the other hydrants the force required to open and close appeared about normal.

TABLE VI.—CONSTRUCTION AND STRENGTH OF HYDRANTS.

1	2	3	4		5		6		7
NAME OF HYDRANT.	Weight of Hydrant.	Inside Volume of Hydrant.	Time Required to Drain.		HYDROSTATIC TEST OF MAIN VALVE.		HYDROSTATIC TEST OF HYDRANT BARREL.		Result of Test of Valve Stems.
					Pressure Put on.	Result.	Pressure put on.	Result.	
	lbs.	cu. ft.	m.	s.	lbs.		lbs.		
Beaumont	290	0.69	2	30	180	Valve tight.	300	Drip leaked very little	Stems twisted off.
Chapman No. 1	435	1.39	7	30	180	Pressure dropped in ½ min. 80 lbs.	150	Caps leaked.	Stem twisted out of shape.
" No. 2	425	1.44	7	15					
" No. 3	734	2.29	5	11					
Chapman 3-way	577	1.63	7	01					
" 4-way	5?0	2.24							
Coffin Gate	467	1.59	....	.....	180	Pressure dropped in 1 min. 80 lbs.	240	Caps leaked.	Not injured.
" Compression	506	1.44	7	30	180	Pressure dropped in 1 min. 75 lbs.	280	" "	Stem strained, but not broken.
Corey 4-in	418	1.55	2	30	180	Pressure dropped in 1 min. 80 lbs.	200	" "	
" 5-in	560	2.24	3	40	180	Pressure dropped in 40 secs. 80 lbs.	300	" "	Not injured.
Glamorgan 4-in	531	1.22	2	05	180	Valve tight.	280	" "	Top of stem broken off.
Holyoke Gate, 4-in	256	1.02							
Holyoke Gate, 6½-in	500	2.16	9	00	180	Pressure dropped in 1 min. 80 lbs.	300	Caps leaked.	Stem twisted off.
" Compression No. 1	669	1.20	4	30	180	Pressure dropped in 1 min. 80 lbs.	220	" "	Bottom of hydrant pushed out.
" " No. 2	538	0.77							
" Gate, 4-way	580	2.24							
" " 6-way	.....	.....	31	06					
Ludlow	497	1.87	3	40	180		280	Caps leaked.	Stem twisted out of shape.
Mathews 5-in	750	1.42	5	30	180	Pressure dropped in 50 secs. 80 lbs.	280	Flange at bottom leaked	Top of stem broken off.
" 4-way	757	1.50							
Pratt & Cady	6?8	2.28	15	00	70	Valve leaked badly.	220	Caps leaked.	Yoke in bottom of hydrant broken.

valves. Two of the hydrants had not positive drips, and in both of these the drips got out of order and failed to close when the main valve was open. The importance of having an absolutely reliable drip-valve, so as to reduce the chances of freezing to a minimum, must be apparent to every one.

Columns 5 and 6.—The hydrostatic tests were made with an ordinary high-pressure hand pump, which is shown at one corner of the room in Fig. 5. The aim was not to break anything, but to see how the hydrant would act with a considerable increase of pressure above the normal. Table VI. shows the results in full.

Column 7.—A broken or twisted valve stem is not an uncommon result of the excitement of a large fire. Sometimes a hydrant sticks, or a mistake is made in the direction to turn for opening or closing. Often in such cases, if one man cannot start the hydrant, two or more men take hold. In this connection it is somewhat surprising that so far as known no one has yet made a hydrant which is not liable to serious damage if a forcible attempt is made to turn it in the wrong direction, either opening or closing; something accomplishing what the ratchet on a stem-winding watch accomplishes seems possible and greatly preferable to the simple limit of breaking strength. Such a device might also give the fireman immediate evidence that he was wrong, thus saving time. A hydrant cannot be expected to stand unlimited abuse, but to get some idea of the ability of the hydrants to withstand such usage the following tests were made: Two ordinary men were instructed to open each hydrant, using the regular hydrant wrench, which is 17 inches long, exerting their maximum strength in an effort to open the hydrant beyond its natural limit. If no injury resulted they closed the hydrant, exerting again their maximum strength after the hydrant was completely shut.

These tests therefore took the strength of two ordinary men using a wrench of definite length as the measure of the force applied. It is not exact in any way, but gives some results which, in a practical way, are somewhat useful.

In most instances some injury was done to the hydrant, the stem generally being the point to give way, though in one case the bottom of a hydrant was actually pushed out by attempting to open it beyond its natural limit.

Durability and Repair.—Time did not give a chance for any

complete tests on the durability of the working parts of the hydrant, but starting with the assumption that a hydrant might be opened on an average ten times a year, and should be good for a service of at least twenty years, each hydrant was opened and closed 200 times. No special derangement or wear resulted, except in two cases, in which the stuffing-box nut on top of the hydrants showed a tendency to work loose.

In a number of the hydrants the design has been made with the idea of facilitating repairs. In some cases this resulted in considerable restriction to the waterways. It is believed that in most cases, by keeping the desirability of free waterways in mind, ability to make quick repairs can be combined with smooth, free waterways. In this connection the friction-loss tests will be of value in showing what can be done and what should be avoided.

METER TESTS.

As previously stated, the two large meters used in the open-butt tests were calibrated by the nozzles, the quantities by the nozzles being assumed to be correct. This calibration work was done in connection with the regular tests, but has been tabulated independently, and the data in full are given in Tables VII. and VIII. From these data, a curve was plotted for each meter with nozzle readings for one coördinate, and meter readings for the other (see Fig. 212). From this curve the corrected meter readings were taken directly in working up the open-butt tests.

The arrangement of the apparatus gave an excellent opportunity to determine the obstruction to the flow of water; that is, the friction loss caused by the meters. As this may become an important feature where handling large quantities of water, and as data on large meters with high rates of flow are not very complete, a few tests were made to determine this loss. The method of testing followed was similar to that employed in the series of meter tests reported by Mr. E. V. French in the *Journal* of the New England Water-Works Association.* A gauge connection was tapped into the 6-inch pipe at each side of the meter and a short distance from it, the connection being arranged the same as those at the inlet pipe of the hydrants. Between these two connections a mercury U-gauge was attached, and the friction

* "Losses of Pressure Caused by Meters in Factory Fire Supplies," *Journal* of the New England Water-Works Association, Vol. XXII., No. 2.

loss caused by the meter with various rates of flow was read directly from this gauge. The loss in the short lengths of the 6-inch pipe between the gauge connections and the meter is included in the meter loss, as it was too small to be of practical

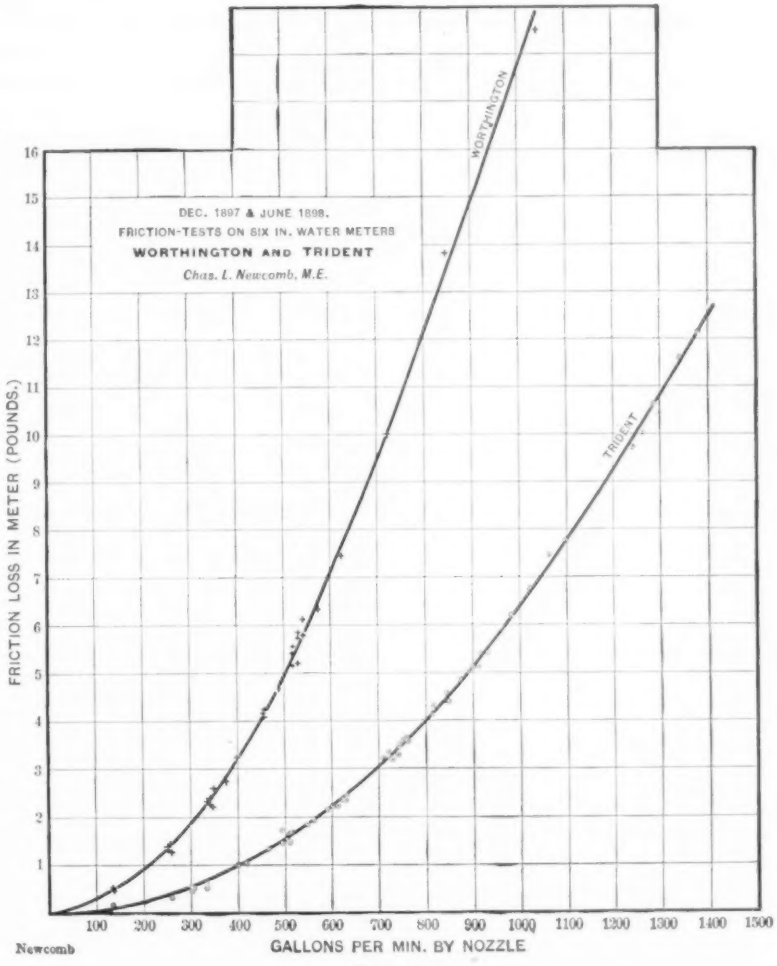


FIG. 212.

importance. This data also appear in full in Tables VII. and VIII., and the results have been plotted on Fig. 213, with the gallons per minute as one coördinate, and the losses in pressure in pounds as the other.

The question is sometimes raised as to what would happen if the moving parts of a meter became blocked, an accident which may happen where a fish-trap is not provided with the meter, and which is a possibility under some conditions even where a fish-trap is used. After the hydrant tests were completed, a few experiments in this line were made with the Neptune meter, the owners having kindly consented that any sort of tests should be made with it, regardless of the results. The cover was removed and wooden wedges inserted, so as to hold the disc in various positions. It was found that it could be blocked so that almost

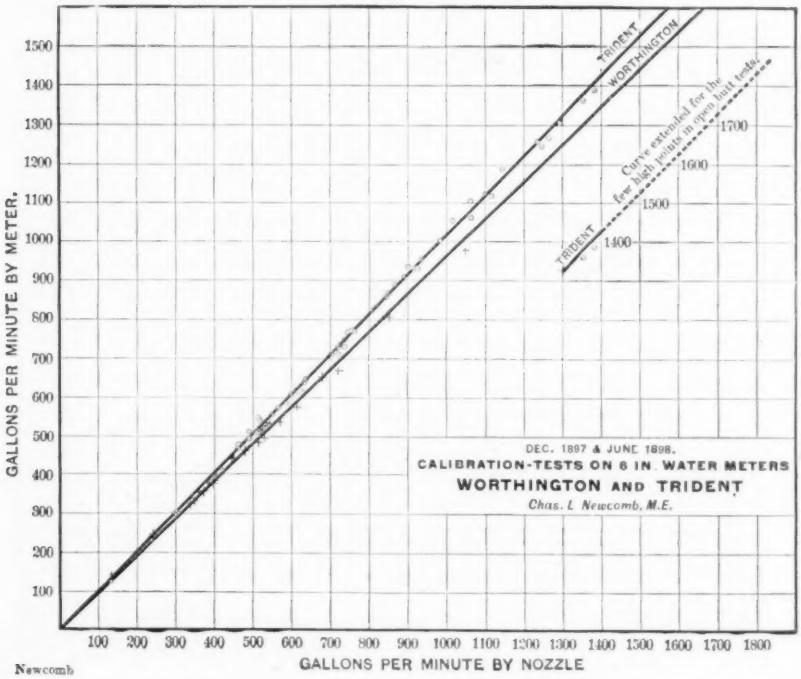


FIG. 213.

no water could go through it, and that any degree of obstruction between this and a free opening could be obtained. With the meter blocked in various positions, full water pressure was turned on behind it, to see if anything would be broken or sprung; and when in the position where practically no water got through the meter, the pressure warped the disc, though not enough to appreciably increase the flow of water.

TABLE VII.

Tests on 6-inch Worthington Water Meter.

1	2	3		4	5	6	7	8	9	10
Test No.	Date of Test.	Duration of Test.		Diam. of Nozzle. Inches.	Pressure at Nozzle. Corrected Pounds.	Gallons per Minute by Nozzle.	Total Cubic Feet by Meter.	Gallons per Minute by Meter.	Diff. in Gals. per Min. Nozzle as Standard (6—8).	P'ds Loss between Gauge Con'ctions eq'ls Meter Loss.
		m.	s.							
488	Dec. 10, 1897	11	01 5/8	3/4	66.70	135	200	135	0	
489	"	11	44 1/5	1 1/8	46.49	255	400	255	0	1.29
490	"	11	29	1 1/4	53.18	340	500	326	+14	2.22
491	"	10	01 1/2	1 1/2	24.14	458	600	448	+10	4.24
492	"	10	11 3/5	2	17.44	518	700	514	+ 4	5.41
493	"	10	24	1 1/4	64.60	375	500	360	15	2.75
494	"	10	42 3/5	1 3/8	30.34	514	700	488	26	5.18
495	"	10	08 1/5	1 1/2	44.63	623	800	590	33	7.45
496	"	10	55 3/5	2	33.84	722	1,000	684	38	10.00
497	"	10	12 3/5	2 1/8	16.87	848	1,100	806	42	13.80
498	"	10	40 3/5	2 1/4	25.53	1,044	1,400	981	63	18.50
499	"	11	12	3/4	65.66	134	200	134	0	0.50
500	"	11	52 1/3	1 1/8	46.07	254	400	252	2	1.31
501	Dec. 11, 1897	11	38 2/5	1 1/4	51.77	336	500	321	15	2.32
502	"	13	26 3/5	1 3/8	23.87	457	800	445	12	4.17
503	"	10	12 3/5	2	17.23	516	700	512	4	5.55
509	"	11	11	3/4	65.68	134	200	134	0	0.50
510	"	11	53 1/5	1 1/8	45.73	254	400	252	2	1.40
511	"	11	32 4/5	1 1/4	52.96	340	500	323	7	2.29
512	"	10	06 2/5	1 3/8	23.95	457	600	444	13	4.18
513	"	10	02	2	18.36	531	700	522	9	5.85
519	Dec. 13, 1897	11	11	3/4	65.7	134	200	134	0	0.50
520	"	11	49 1/2	1 1/8	45.94	254	400	253	1	1.41
521	"	11	30 2/5	1 1/4	53.25	340	500	324	16	2.39
522	"	11	47 1/5	1 3/8	23.65	455	700	444	11	4.18
523	"	11	28 1/5	2	18.3	530	800	522	8	5.75
524	"	11	15	3/4	64.78	133	200	133	0	.54
525	"	11	52 1/5	1 1/8	46.00	254	400	252	2	1.45
526	"	11	33	1 1/4	53.08	340	500	324	16	2.26
527	"	10	08 3/5	1 3/8	23.86	456	600	443	13	4.08
528	"	11	18 3/5	2	18.69	537	800	529	8	5.80
591	Dec. 16, 1897	11	19	2	18.87	540	800	529	11	6.11
592	"	11	01 1/2	1 3/8	14.00	350	500	341	9	18.60
593	"	11	34 1/5	1 1/2	18.2	398	600	386	12	3.24
594	"	11	22 4/5	1 3/4	26.9	485	700	462	23	4.65
595	"	10	28 4/5	1 7/8	32.0	528	700	498	30	5.20
596	"	11	06 1/2	1 7/8	37.2	570	800	539	31	6.31
609	"	5	56	2	9.76	388	300	378	10	.....
610	"	5	45 3/5	2	29.76	578	500	650	28	.....
611	"	11	45	1 3/8	18.23	398	600	382	16	.....
612	"	11	21 3/5	1 3/4	27.04	485	700	461	24	.....

CURVES AND DATA SHEETS.

The curves and data sheets for all tests are arranged in the order given below:

Plates I. to XXII. (Figs. 162 to 183, inclusive).—Friction losses in the barrels, and in the hydrant as a whole, with various rates of flow.

Plates XXIII. to XXV. (Figs. 184, 185 and 186).—Discharging capacity of open butts for varying pressures at the top of the hydrant barrels.

Data Sheets for All Tests.—These sheets give the fundamental data of the tests and the main steps of the computations. The friction-loss tests are arranged in alphabetical order in accordance with the names of the hydrants; the open-butt tests follow and are arranged in order of test numbers, *i.e.*, chronologically.

TABLE VIII.

TEST ON 6-INCH TRIDENT WATER METER.

1	2	3	4	5	6	7	8	9	10
Test No.	Date of Test.	Durat'n of Test.	Diam. of Nozzle. Inches.	Pres're at Nozzle Cor'ted Lbs.	Gallons per Minute by Nozzle.	Total Cubic Feet by Meter.	Gallons per Min. by Meter.	Diff. in Gals. per Min. Nozzle as Standard (6-8).	Pounds Loss between Gauge Connections Equals Meter Loss.
		m. s.							
875	June 2, 1898.	5 05	3/4	66.04	135	90	130	+ 5	.....
876	" " "	5 02	1	44.41	250	170	247	+ 3	.....
877	" " "	5 03	1	50.31	332	230	328	+ 4	.....
878	" " "	5 01	1	24.52	463	320	471	− 8	.....
879	" " "	5 00	2	16.93	511	360	530	− 19	1.68
880	" " "	4 59	1	41.62	301	200	296	+ 5	0.47
881	" " "	4 57	1	29.53	507	340	507	0	1.50
882	" " "	4 59	1	44.52	623	420	619	+ 4	2.32
883	" " "	5 01	2	33.26	708	490	723	− 15	3.20
884	" " "	5 01	2	15.52	807	560	826	− 19	4.10
885	" " "	1 59	2	19.60	916	250	929	− 13	5.40
886	" " "	1 59	2	35.35	738	200	741	− 3	3.30
887	" " "	1 59	1	39.99	590	160	592	− 2	2.16
888	" " "	2 59	1	18.43	400	160	395	− 5	1.02
889	" " "	2 12	1	41.52	300	90	301	− 1	0.57
890	" " "	2 02	1	29.29	506	140	507	− 1	1.53
891	" " "	2 02	1	43.00	612	170	615	− 3	2.21
892	" " "	2 02	2	33.56	720	200	727	− 7	3.23
893	" " "	2 02	2	15.72	819	230	835	− 16	4.30
894	" " "	2 05	2	19.47	549	160	561	− 12	1.88
895	" " "	2 05	2	34.74	731	210	745	− 14	3.46
896	" " "	2 05	2	17.85	873	250	893	− 20	4.89
897	" " "	2 00	2	22.65	982	270	1,001	− 19	6.18
898	" " "	2 31	2 & 1	23.78 24.12	752	260	772	− 20	3.57
899	" " "	1 31	2 & 1	34.05 34.86	898	190	929	− 31	5.10
900	" " "	1 33	2½ & 1	18.32 22.07	1,063	230	1,104	− 41	7.49
901	" " "	1 30	2½ & 1¼	20.67 21.25	1,156	240	1,194	− 38	8.77
902	" " "	2 29	2½ 1¼ & 1	17.50 24.42	1,237	330	1,260	− 23	9.67
903	" " "	2 30	2½ 1¼ & 1¼	13.98 19.89	1,109	380	1,126	− 17	8.29
904	" " "	1 32	2½ 1¼ & 1¼	.....	1,277	270	1,315	− 18	10.90
1,197	June 21, 1898.	3 18	¾	65.31	134	60	136	− 2	0.14
1,198	" " "	3 09	1	47.16	258	110	260	− 2	0.32
1,199	" " "	3 05	1	51.51	334	140	338	− 4	0.66
1,200	" " "	3 07	1	24.61	463	200	478	− 15	1.33
1,201	" " "	3 05	2	16.05	507	220	534	− 27	1.66
1,202	" " "	3 13	1	41.47	302	130	302	0	0.55
1,203	" " "	3 05	1	28.34	494	210	508	− 14	1.47
1,204	" " "	3 02	1	45.36	629	260	639	− 10	2.31
1,205	" " "	3 02	2	34.06	725	300	740	− 15	3.20
1,206	" " "	3 02	2	16.59	843	350	861	− 18	4.58
1,207	" " "	3 03	1	19.14	417	170	416	+ 1	1.05
1,208	" " "	3 05	1	41.10	600	170	608	− 8	2.22
1,209	" " "	3 05	2	37.63	761	320	775	− 14	3.57
1,210	" " "	3 04	2	20.21	930	390	949	− 19	5.40
1,211	" " "	3 07	2	20.44	561	240	574	− 13	1.97
1,212	" " "	3 03	2	35.87	745	310	759	− 14	3.49
1,213	" " "	3 03	2	18.49	889	370	906	− 17	5.02
1,214	" " "	3 01	2	25.53	1,023	430	1,063	− 40	6.77
1,215	" " "	3 14	1	42.27	307	130	301	+ 6	0.56
1,216	" " "	3 00	1	27.99	494	200	498	− 4	1.73
1,217	" " "	3 01	1	45.59	630	260	642	− 12	2.44
1,218	" " "	3 03	2	33.61	720	300	732	− 12	3.30
1,219	" " "	3 01	2	16.86	849	350	868	− 19	4.40
1,220	" " "	2 30	2 & 1	23.83 26.32	759	260	775	− 16	3.60
1,221	" " "	2 24	2 & 1	34.23 38.33	910	320	929	− 19	5.13
1,222	" " "	2 31	2 & 1¼	19.11 27.30	1,100	380	1,126	− 26	7.77
1,223	" " "	2 33	2 & 1¼	22.86 29.22	1,243	430	1,256	− 13	9.70
1,224	" " "	2 31	2 & 1¼	22.31 24.95	1,364	430	1,275	− 11	10.01
1,225	" " "	2 31	2½ 1¼ & 1¼	18.91 26.90	1,392	440	1,308	− 16	10.60
1,226	" " "	2 31	2½ 1¼ & 1¼	20.56 29.57	1,346	460	1,364	− 18	11.56
1,227	" " "	2 31	2½ 1¼ & 1¼	19.56 25.32	1,387	470	1,391	− 4	12.13
1,228	" " "	2 30	2½ 1¼ & 1¼	11.45 15.25	1,065	360	1,073	− 8	7.47

EXPERIMENTS ON BEAUMONT 4-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 38° Fahr. **Temperature of Atmosphere, 44° Fahr.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)−(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Frict'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd Bbl. (9)−[(15)−(12)].	From 6″ to 2½″ Piez'rs equals the Total Loss. (8)−(17)−(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)−(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Dec. 10.																											
	m s.		in.	g'e.	cor.		g'ge.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
488	11 01	1	¾	67.1	66.6	135	65.7	69.0	0.90	0.84	0.30	0.18	0.66	1.53	0.03	0.01		3.44	0.19	0.08	8.83	1.20	0.52	0.11	0.33	0.22	67	
489	11 44	1	1¼	46.9	46.4	256	53.76	57.1	3.40	3.16	0.67	0.62	2.54	2.90	0.13	0.06		6.56	0.66	0.29	16.72	4.35	1.89	0.39	1.33	0.94	71	
490	11 29	1	1½	53.6	53.1	340	67.4	70.7	5.76	5.35	1.05	0.98	4.37	3.86	0.23	0.10		8.70	1.17	0.51	22.20	7.66	3.33	0.57	2.12	1.75	73	
491	10 01½	1	1¾	24.5	24.1	459	58.4	61.7	10.99	10.20	2.01	1.90	8.30	5.21	0.43	0.19		11.76	2.14	0.93	30.00	14.00	6.07	1.16	4.32	3.16	73	
492	10 11½	1	2	17.8	17.4	518	62.3	65.6	14.55	13.50	2.69	2.51	10.99	5.88	0.54	0.23		13.26	2.73	1.18	33.86	17.82	7.73	1.56	6.00	4.44	74	
493	10 24	2	1¼	65.0	64.5	374	65.8	69.1	2.10	1.92	1.29	1.20	0.72	4.23	0.28	0.12	Area = 12.59 or Area of 4″. ⊙	9.57	1.42	0.61	12.22	2.32	1.02	0.71	1.02	0.31	30	R. H. Outlet by deduction.
494	10 42	2	1½	30.7	30.3	514	39.3	42.6	4.12	3.83	2.50	2.33	1.50	5.84	0.53	0.23		13.15	2.68	1.16	16.77	4.39	1.91	1.40	2.15	0.75	35	
495	10 08	2	1¾	45.0	44.6	623	57.8	61.1	6.08	5.67	3.70	3.45	2.22	7.07	0.78	0.34		15.94	3.93	1.71	20.34	6.43	2.80	2.08	3.21	1.13	35	
496	10 55	2	2	34.2	33.8	721	53.6	56.9	8.11	7.53	4.99	4.62	2.91	8.19	1.04	0.45		18.41	5.27	2.41	23.52	8.61	3.74	2.66	4.24	1.58	37	
497	10 12	2	2½	17.3	16.8	848	47.8	51.1	11.49	10.68	7.03	6.54	4.14	9.64	1.43	0.62		21.66	7.28	3.29	27.68	11.94	5.18	3.87	6.12	2.25	37	

EXPERIMENTS ON BEAUMONT 4-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. **Temperature of Atmosphere, 58°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entr'ce. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydrant Bbl. at Gauge Connection. Area = 12.59 or Area of 4″. ⊙	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fric't'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—[(15)—(12)].	Act'l Fric't'n Losses. From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—[(17)—(12)].	Act'l Fric't'n Losses. From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Dec. 11.																											
	m. s.		in.	ga'ge.	corr.		g'e.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
499	11 12	1	⅝	66.06	65.61	134	64.9	68.2	0.92	0.86	0.2	0.19	0.67	1.52	0.03	0.01		3.42	0.20	0.09	8.77	1.18	0.51	0.11		0.25	70	
500	11 52½	1	1	46.47	46.02	254	55.8	59.1	3.4	3.17	0.6	0.55	2.62	2.88	0.13	0.06		6.53	0.65	0.28	16.58	4.28	1.87	0.33	1.36	1.03	76	
501	11 38⅔	1	1¼	52.17	51.72	335	65.5	68.8	5.6	5.32	1.03	0.96	4.36	3.80	0.22	0.10		8.57	1.13	0.49	21.87	7.43	3.24	0.57	2.08	1.51	73	
502	13 26⅔	1	1½	24.27	23.85	456	68.8	72.1	10.87	10.07	2.00	1.88	8.19	5.16	0.42	0.18		11.65	2.11	0.91	29.79	13.80	5 98	1.14	4.27	3.13	73	
503	10 12⅔	1	2	17.63	17.21	516	59.6	62.9	14.34	13.30	2.68	2.50	10.80	5.85	0.53	0.23		13.19	2.70	1.17	33.73	17.69	7.67	1.56	5.86	4.30	73	
	Dec. 11.																											
504	10 28	2	1¼	61.48	64.03	373	65.2	68.5	2.12	1.99	1.3	1.20	0.79	4.23	0.28	0.12		9.55	1.41	0.61	12.19	2.30	1.00	0.71	1.11	0.40	36	
505	10 36	2	1⅜	30.45	30.03	512	38.3	41.6	4.11	3.84	2.53	2.37	1.47	5.80	0.53	0.23		13.08	2.66	1.15	16.72	4.35	1.90	1.45	2.17	0.72	33	
506	10 10⅝	2	1½	44.84	44.42	621	57.7	61.0	5.99	5.55	3.7	3.45	2.10	7.05	0.77	0.33		15.88	3.90	1.70	20.27	6.38	2.78	2.08	3.10	1.02	33	
507	11 02	2	2	33.6	33 18	708	53.5	56.8	8.18	7.61	4.96	4.59	3.02	8.04	1.00	0.43		18.09	5.08	2.21	33.10	8.29	3.60	2.81	4.44	1.63	37	
508	10 25	2	2¼	16.72	16.30	833	56.4	59.7	11.28	10.49	6.84	6.36	4.13	9.46	1.39	0.60		21.28	7.03	3.08	27.18	11.52	5.01	3.88	6.08	2.20	36	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON BEAUMONT 4-INCH HYDRANT.

(With Trident Meter.)

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, {32° = 23d. / 15° = 24th.

1	2	3	4	5		6	7		8		9		10	11	12		13	14		15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entr'nce (Lbs.).		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydrant Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diamet'r of Hydrant Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.		Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresp'd'g to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—[(15)—(12)].	Actual Friction Losses. From 6″ to 2½″ Piez'r Equals the Total Loss. (8)—(17)—(11).	Actual Friction Losses. From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss (20) × 100 ÷ (19).	Remarks.
	Dec. 24.																											
	m. | s.		in.	g'ge.	cor.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.		ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.	
702	5 | 33	1	¾	67.95	65.50	134	64.7	68.0	1.0	0.94	0.2	0.19	0.75	1.52	0.03	0.01		3.42	0.20	0.09	8.77	1.18	0.51	0.11	0.44	0.33	75	
703	5 | 43½	1	1¼	46.21	45.76	254	52.0	55.3	3.6	3.37	0.6	0.56	2.81	2.88	0.13	0.06		6.53	0.65	0.28	16.58	4.28	1.85	0.34	1.56	1.42	78	
704	6 | 35½	1	1½	53.34	52.89	339	66.0	69.3	6.1	5.68	1.0	0.94	4.74	3.84	0.23	0.10		8.68	1.16	0.50	22.12	7.61	3.32	0.54	2.46	1.92	78	
705	6 | 25	1	1¾	24.22	23.80	456	53.0	56.3	11.5	10.70	1.9	1.76	8.94	5.16	0.42	0.18		11.65	2.11	0.91	29.79	13.80	5.98	1.03	4.90	3.87	79	
706	5 | 18	1	2	19.4	18.98	542	63.4	66.7	16.76	15.54	2.8	2.60	12.94	6.14	0.50	0.25		13.84	2.97	1.29	35.41	19.48	8.46	1.56	7.33	5.77	79	
	Dec. 23.																											
696	5 | 56½	2	1½	65.81	65.36	377	67.4	70.7	2.2	2.04	1.3	1.20	0.84	4.26	0.28	0.12		9.63	1.43	0.62	12.32	2.34	1.01	0.70	1.15	0.45	39	
697	5 | 49½	2	1¾	30.47	30.05	512	37.7	41.0	4.06	3.79	2.5	2.33	1.46	5.80	0.53	0.23		13.08	2.66	1.15	16.72	4.35	1.90	1.41	2.12	0.71	33	
698	5 | 56½	2	1¾	45.26	44.84	624	57.8	61.1	6.12	5.69	3.58	2.34	2.35	7.09	0.78	0.34		15.97	3.95	1.72	20.39	6.46	2.82	1.96	3.21	1.25	39	
699	5 | 04½	2	2	34.74	34.32	728	54.0	57.3	8.32	7.73	4.95	4.59	3.14	8.27	1.06	0.46		18.60	5.38	2.34	23.75	8.78	3.81	2.71	4.38	1.67	38	
700	5 | 12	2	2½	17.28	16.86	850	46.0	49.3	11.55	10.73	6.77	6.28	4.45	9.64	1.45	0.63		21.70	7.32	3.19	27.73	12.00	5.21	3.72	6.15	2.43	40	

Area = 12.50 or Area of 4″. ⊙

L. H. Outlets by deduction.

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON BEAUMONT 4-INCH HYDRANT.

(LEFT-HAND OUTLET.)

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 38°. **Temperature of Atmosphere, 22°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6" Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6" Piezometer and 2½" Piezometer. (Lbs.)		Measured Loss between 6" Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½" Piezom'r. (8)−(9).	Velocity at 6" Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½" Piezometer.	Head Correspond'g to Column 16.		Actʼl Frictʼn Losses. From 6" Piezʼr to Gauge Connectʼn at Top of Hyd. Bbl. (9)−[15)−(12)].	From 6" to 2½" Piezʼrs Equals the Total Loss. (8)−[(17)−(12)].	From Top of Hyd. Bbl. to 2½" Piezʼr. (19)−(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Dec. 29.																											
	m. s.		in.	g'ge.	cor.		g'e.	cor.	g'ge.	corr.	g'e	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
765	5 33⅔	1	¾	65.84	65.39	134	64.0	67.3	1.0	0.93	.15	0.14	0.79	1.52	0.03	0.01		3.42	0.18	0.08	8.77	1.18	0.51	0.07	0.48	0.36	84	
766	5 45⅔	1	1¼	46.1	45.65	253	52.5	55.8	3.6	3.36	.55	0.51	2.85	2.87	0.13	0.06		6.48	0.64	0.28	16.50	4.24	1.84	0.29	1.58	1.29	81	
767	6 43	1	1¼	51.55	51.10	334	64.3	67.6	5.91	5.49	.9	0.84	4.65	3.78	0.29	0.13		8.55	1.12	0.49	21.87	7.40	3.23	0.48	2.39	1.91	80	
768	6 22⅔	1	1½	24.5	24.08	459	54.6	57.9	11.54	10.73	1.8	1.67	9.06	5.21	0.43	0.19		11.76	2.14	0.93	30.00	14.00	6.07	0.93	4.85	3.92	81	
769	5 30⅔	1	2	18.03	17.61	521	61.5	64.8	15.7	14.58	2.47	2.27	12.31	5.92	0.55	0.24		13.32	2.75	1.19	34.04	18.02	7.82	1.32	7.00	5.68	81	
770	5 35	1	1¼	64.48	64.03	130	64.0	67.3	1.0	0.93	.15	0.14	0.79	1.48	0.03	0.01	Area, 12.59 or Area of 4". ⊙	3.31	0.17	0.07	8.50	1.11	0.48	0.08	0.46	0.38	83	
771	5 47⅔	1	1⅛	45.87	45.42	252	52.0	55.3	3.6	3.36	.55	0.51	2.85	2.83	0.12	0.05		6.44	0.63	0.27	16.44	4.20	1.83	0.29	1.58	1.29	81	
772	6 40⅔	1	1¼	52.61	52.16	337	65.8	69.1	5.95	5.51	.9	0.84	4.67	3.82	0.23	0.10		8.61	1.13	0.49	22.00	7.52	3.27	0.45	2.34	1.89	81	
773	5 24⅔	1	1¾	24.08	23.66	454	53.7	56.8	11.45	10.63	1.8	1.67	8.96	5.15	0.42	0.18		11.61	2.09	0.91	29.65	13.69	5.94	0.94	4.87	3.93	81	
774	5 22⅔	1	2	18.76	18.34	530	63.4	66.7	16.43	15.26	2.5	2.33	12.93	6.01	0.56	0.24		13.56	2.85	1.24	34.62	18.63	8.08	1.33	7.42	6.09	82	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON BEAUMONT 4-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. **Temperature of Atmosphere, 22°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entr'ce. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fricт'n Losses. From 6″ Piez'r to Gauge Connection at top of Hyd. Bbl. (9)—[(15)+(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—[(17)—(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Dec. 20. m. s.		in.	ga'ge.	cor.		g'e.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
775	5 34½	1	1 1/16	66.23	65.78	134	64.5	67.8	0.97	0.90	0.15	0.14	0.76	1.52	0.03	0.01		3.42	0.18	0.08	8.77	1.18	0.51	0.07	0.40	0.33	82	
776	5 46	1	1 1/16	45.99	45.54	253	52.2	55.5	0.34	3.16	0.60	0.56	2.60	2.87	0.13	0.06		6.48	0.64	0.28	16.50	4.24	1.84	0.34	1.38	1.04	73	
777	6 36	1	1 1/16	53.3	52.85	339	66.7	70.0	5.75	5.34	1.05	0.98	4.36	3.84	0.23	0.10		8.68	1.16	0.50	22.12	7.61	3.32	0.58	2.12	1.54	74	
778	6 23½	1	1 1/16	24.5	24.06	459	56.1	59.4	10 88	10.10	2.00	1.87	8.23	5.21	0.43	0.19		11.76	1.14	0.93	30.00	14.00	6.07	1.13	4.32	3.09	73	
779	5 26½	1	2	18.31	17.87	526	63.4	66.7	14.86	13.79	2.77	2.56	11.23	5.97	0.55	0.24		13.44	2.80	1.21	34.38	18.37	7.97	1.59	6.06	4.47	74	
780	5 36½	1	1 1/16	65.71	65.26	134	64.2	67.5	0.96	0.90	0.15	0.14	0.76	1.52	0.03	0.01	Area = 12.59, or Area of 4″ ©.	3.42	0.18	0.08	8.77	1.17	0.51	0.07	0.40	0.33	82	
781	5 43	1	1 1/16	46.35	45.90	254	53.0	56.3	3.04	3.16	0.06	0.56	2.60	2.88	0.13	0.06		6.53	0.65	0.28	16.58	4.28	1.87	0.34	1.35	1.01	75	
782	6 35½	1	1 1/16	53.1	52.65	338	66.8	70.1	5.76	5.35	1.07	1.00	4.35	3.83	0.23	0.10		8.65	1.15	0.50	22.06	7.57	3.29	0.60	2.16	1.56	73	
783	6 24½	1	1 1/16	24.21	23.79	456	55.0	58.3	10.84	10.06	2.00	1.87	8.19	5.16	0.42	0.18		11.65	2.11	0.91	29.79	13.80	5.98	1.14	4.26	3.12	73	
784	5 22½	1	2	18.76	18.34	531	65.2	68.5	15.34	14.26	2.83	2.63	11.63	6.02	0.56	0.24		13.58	2.86	1.24	34.70	18.72	8.12	1.63	6.38	4.75	74	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON CHAPMAN GATE (No. 1) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 44°. Temperature of Atmosphere, 45°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Test.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Frict'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (6)—[(15)—(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Nov. 11, 1897.		in.	g'e.	cor.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
48	10 Min.	1	⅝	15.9	65.5	134	65.3	68.6	0.90	0.87	0.05	0.05	0.82	1.52	0.03	0.01	4.75	2.42	0.09	0.04	8.76	1.18	0.52	0.02	0.36	0.34	94	
49	"	1	1	46.3	44.9	252	52.5	55.8	3.30	3.06	0.20	0.19	2.87	2.84	0.12	0.05	"	4.53	0.32	0.14	16.45	4.21	1.83	0.10	1.28	1.18	92	
51	"	1	1¼	53.7	52.3	338	67.0	70.3	5.57	5.18	0.40	0.37	4.81	3.84	0.23	0.10	"	6.12	0.58	0.25	22.06	7.57	3.29	0.22	1.99	1.77	89	
52	"	1	1¼	23.9	22.7	445	54.1	57.4	10.27	9.55	0.75	0.70	8.85	5.03	0.40	0.17	"	8.02	1.00	0.43	29.05	13.16	5.72	0.44	4.00	3.56	89	
54	"	1	2	17.2	16.0	497	56.8	60.1	13.25	12.39	0.90	0.84	11.45	5.63	0.49	0.21	"	8.98	1.25	0.54	32.50	16.38	7.11	0.51	5.39	4.88	91	
55	"	2	1¼	62.9	61.5	366	65.7	69.0	2.20	2.03	0.35	0.33	1.70	4.14	0.27	0.12	"	6.60	0.68	0.29	11 97	2.21	0.97	0.16	1.18	1.02	86	
56	"	2	1½	30.7	29.5	508	39.8	43.1	4.31	4.00	0.75	0.70	3.30	5.77	0.52	0.22	"	9.20	1.32	0.57	16.58	4.28	1.87	0.35	2.35	2.00	85	
57	"	2	1½	44.4	43.2	614	58.2	61.5	6.15	5.72	1.00	0.93	4.79	6.98	0.76	0.33	"	11 13	1.93	0.84	20.03	6.24	2.72	0.42	3.33	2.91	87	
58	"	2	2	33.8	32.7	710	53.4	56.7	8.29	7.70	1.40	1.29	6.41	8.06	1.01	0.44	"	12.85	2.57	1.11	23.16	8.33	3.62	0.62	4.52	3.90	86	
	Nov. 12, 1897.																											
59	.....	2	2½	16.4	15.2	809	46.0	49.3	11.24	10.45	1.80	1.67	8 78	9.20	1.31	0.57	"	14.67	3.35	1.45	26.40	10.87	4.72	0.79	6.30	5.51	87	

EXPERIMENTS ON CHAPMAN GATE (No. 1) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 42°. Temperature of Atmosphere, 35°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17	18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.	Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—(15)—(12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Nov. 16, 1897.		in.	g'e.	cor.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.	
169	10 Min	1	⅝	66.8	65.5	134	65.2	68.5	1.00	0.90	0.05	0.05	0.85	1.51	0.03	0.01	4.75	2.41	0.09	0.04	8.76	1.18	0.51	0.02	0.40	0.38	95	
170	"	1	1	46.7	45.4	257	52.4	55.7	3.47	3.30	0.20	0.18	3.02	2.91	0.13	0.06	"	4.64	0.33	0.14	16.78	4.38	1.90	0.10	1.36	1.26	93	
171	"	1	1⅛	52.9	51.6	335	65.2	68.5	5.86	5.46	0.40	0.37	5.09	3.79	0.22	0.10	"	6.04	0.57	0.25	21.87	7.44	3.24	0.22	2.32	2.10	91	
172	"	1	1⅛	24.3	23.1	450	52.1	55.4	10.90	10.12	0.70	0.65	9.47	5.10	0.41	0.18	"	8.13	1.03	0.45	29.40	13.45	5.83	0.38	4.47	4.09	91	
173	"	1	2	19.1	17.9	524	62.7	66.0	15.56	14.46	1.00	0.93	13.53	5.95	0.55	0.24	"	9.49	1.40	0.61	34.23	18.23	7.92	0.56	6.78	6.22	92	
174	"	2	1¼	64.3	63.0	370	65.0	68.3	2.30	2.14	0.40	0.37	1.77	4.19	0.27	0.12	"	6.68	0.69	0.30	12.08	2.27	0.98	0.19	1.28	1.09	85	
175	"	2	1⅜	44.7	43.5	616	57.7	61.0	6.30	5.87	1.00	0.93	4.94	7.00	0.76	0.33	"	11.16	1.94	0.84	20.10	6.28	2.73	0.42	3.47	3.05	88	
176	"	2	1½	30.3	29.1	504	37.5	40.8	4.30	4.00	0.70	0.65	3.35	5.73	0.51	0.22	"	9.14	1.30	0.56	16.47	4.22	1.83	0.31	2.39	2.08	87	
177	"	2	2	34.0	32.8	710	52.9	56.2	8.49	7.90	1.40	1.30	6.69	8.06	1.01	0.44	"	12.85	2.57	1.11	23.17	8.33	3.62	0.63	4.72	4.09	87	
178	"	2	2½	16.6	15.4	812	43.9	47.2	11.10	10.32	1.93	1.79	8.53	9.23	1.32	0.57	"	14.72	3.37	1.47	26.50	10.96	4.75	0.89	6.14	5.25	86	

EXPERIMENTS ON CHAPMAN TWO-WAY (No. 2) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 44°. Temperature of Atmosphere, 45°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl and 2½″ Piezom'r. (8) − (9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fricт'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9) − (15) − (19).	From 6″ to 1½″ Piez'rs Equals the Total Loss. (8) − (17) − (21).	From Top of Hydr. Bbl. to 2½″ Piez'r. (19) − (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	June 1, 1898.																											
	m.	s.		g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
870	4	59½	2	1¼	42.36	41.91	302.0	No cor't'n for ele't'n	1.15	1.06	0.20	0.19	0.87	3.42	0.18	0.08	5	4.92	0.38	0.16	9.89	1.50	0.66	0.11	0.48	0.37	85	
871	5	01½	2	1¼	29.81	28.86	502.0	....	3.01	2.83	0.57	0.42	2.41	5.70	0.51	0.22	"	8.2	1.05	0.46	16.38	4.19	1.82	0.18	1.23	1.05	85	
872	4	56½	2	1¼	43.63	43.18	614.0	....	4.53	4.22	0.85	0.78	3.44	6.98	0.76	0.33	"	10.05	1.57	0.68	20.03	6.24	2.73	0.43	1.82	1.39	76	
873	5	03½	2	2	32.12	31.67	700.0	57	5.98	5.54	1.15	1.06	4.48	7.95	0.98	0.42	"	11.45	2.04	0.88	22.84	8.10	3.52	0.60	2.44	1.84	75	
874	4	59½	2	2½	16.23	15.78	822.0	....	8.41	7.82	1.56	1.46	6.35	9.53	1.35	0.59	"	13.44	2.81	1.22	26.82	1.22	4.87	0.83	3.54	2.71	77	

Zero corr. = 0.45 for all diameters.

EXPERIMENTS ON CHAPMAN TWO-WAY (No. 2) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 50°. Temperature of Atmosphere, 70°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Test.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actʼl Frictʼn Losses. From 6″ Pizr't to Gauge Connect'n at Top of Hyd. Bbl (9)—(15)—(12).	From 6″ to 2½″ Pizr'rs Equals the Total Loss. (8)—(17)–(12).	From Top of Hyd. Bbl. to 2½″ Pizr'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	June 8, 1898.		in.	ga'ge.	corr.		g'e.	cor.	ga.	rr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
966	3 | 18½	1	¾	62.2	61.77	130	....	60	0.	0.75	0.06	0.08	0.69	1.48	0.03	0.01	5	2.13	0.07	0.03	8.50	1.12	0.49	0.04	0.27	0.23	85	} *
967	3 | 10⅝	1	1¼	46.47	46.04	254	....	56	2.70	4.52	0.24	0.23	2.29	2.88	0.13	0.06	"	4.15	0.25	0.14	16.58	4.28	1.87	0.17	0.71	0.54	76	
968	3 | 04⅝	1	1½	52.12	51.69	335	....	60	4.25	3.99	0.25	0.24	3.75	3.79	0.29	0.13	"	5.46	0.46	0.20	21.86	7.43	3.23	0.17	0.89	0.72	81	} *
	June 9, 1898.																											
969	3 | 12½	1	¾	66.43	65.9	134	....	57.4	0.80	0.75	0.07	0.07	0.68	1.52	0.03	0.01	"	2.19	0.07	0.03	8.62	1.18	0.52	0.05	0.24	0.19	79	
970	3 | 08½	1	1¼	46.84	46.33	255	....	55	2.60	2.43	0.17	0.16	2.27	2.89	0.13	0.06	"	4.15	0.25	0.12	16.63	4.31	1.88	0.10	0.61	0.51	84	
971	3 | 04½	1	1½	52.36	51.85	335	....	68	4.40	4.08	0.29	0.25	3.81	3.79	0.32	0.10	"	5.46	0.46	0.20	21.87	7.43	3.23	0.17	0.95	0.78	82	
972	3 | 02	1	1¾	24.37	23.88	456	....	60.6	8.27	7.68	0.49	0.45	7.23	5.16	0.42	0.18	"	7.45	0.85	0.37	29.79	13.80	5.99	0.26	1.87	1.61	86	
973	3 | 02	1	2	17.84	17.35	517	....	....	10.98	10.20	0.63	0.60	9.60	5.86	0.53	0.23	"	8.44	1.11	0.48	33.79	17.74	7.69	0.35	2.74	2.39	87	
974	3 | 14	2	1¼	42.13	41.62	301	....	43	1.15	1.06	0.21	0.20	0.86	3.40	0.18	0.08	"	4.92	0.38	0.16	9.83	1.49	0.65	0.12	0.49	0.37	76	
975	3 | 08⅝	2	1¼	31.16	30.67	516	....	40.5	3.14	2.94	0.64	0.60	2.34	5.86	0.54	0.23	"	8.44	1.11	0.48	16.83	4.42	1.92	0.35	1.25	0.90	72	
976	3 | 05⅝	2	1½	44.92	44.43	621	....	60.3	4.65	4.33	0.86	0.78	3.55	7.04	0.78	0.34	"	10.14	1.60	0.69	20.27	6.38	2.78	0.43	1.89	1.46	77	
977	3 | 06⅝	2	2	33.21	32.72	710	....	55	6.04	5.63	1.18	1.09	4.54	8.07	1.01	0.44	"	11.62	2.10	0.91	23.17	8.33	3.62	0.62	2.45	1.83	74	
978	3 | 01⅝	2	2½	16.49	16.00	825	....	47.6	8.23	7.65	1.60	1.49	6.16	9.38	1.37	0.59	"	13.51	2.84	1.23	26.99	11.36	4.93	0.85	3.31	2.46	74	

Zero corr. = 0.43, Nos. 966–968.
" 0.51, small diam. nozzle, } Nos. 969–978.
" 0.49, larger " }

* Repeated; see Nos. 969–971.

EXPERIMENTS ON CHAPMAN TWO-WAY (No. 2) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 50°. **Temperature of Atmosphere, 72°.**

1	2	3	4	5		6	7	8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entr'ce. (Lbs.)	Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8)–(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)–(15)–(12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)–(17)–(12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)–(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	June 11, 1898.																										
	m. s.		in.	ga'ge.	cor.		g'e. cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	feet	lbs.	lbs.	lbs.	lbs.		
1055	2 09⅔	1	¾	66.17	65.65	135	....	66.3	0.78	0.72	0.07	0.07	0.65	1.52	0.03	0.01	5	2.19	0.07	0.03	8.82	1.20	0.53	0.05	0.20	0.15		
1055	2 15	1	1⅛	48.79	48.37	261	....	57.3	2.65	2.48	0.20	0.19	2.29	2.95	0.13	0.06	"	4.25	0.28	0.12	17.03	4.53	1.97	0.13	0.57	0.44		
1057	2 11	1	1½	53.00	52.48	338	....	68.8	4.35	4.03	0.32	0.30	3.73	3.83	0.23	0.10	"	5.52	0.47	0.20	22.06	7.57	3.29	0.20	0.84	0.64		
1058	2 04⅘	1	1¾	24.32	23.82	456	....	56.3	8.28	7.70	0.58	0.53	7.17	5.05	0.42	0.18	"	7.43	0.86	0.37	29.79	13.80	5.99	0.34	1.89	1.55		
1059	2 07⅘	1	2	19.02	18.52	534	....	66.0	11.77	10.93	0.77	0.71	10.22	6.06	0.57	0.25	"	8.73	1.18	0.51	34.89	18.91	8.21	0.45	2.97	2.52	85	82 76

Zero corr. = 0.50 and 0.52.

EXPERIMENTS ON CHAPMAN TWO-WAY (No. 3) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 50°. **Temperature of Atmosphere, 72°.**

	1	2	3	4	5		6	7	8		9		10	11	12		13	14	15		16	17	18	19	20	21	
	Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)	Measured Loss between 6″ Piezometer and 2″ Piezom't'r (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydrant Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydrant Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 14.	Act'l Frict'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—[(15)—(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—[(17)—(12)]	From Top of Hyd. Bbl. to 2½″ Piez'r. [(19)—(18)].	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
		1898. June 11.																									
		m. s.		in.	g'ge.	corr.		g'e. cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.		
Outlet No. 2.	990	3 20	1	¾	65.92	65.40	134	.... 6.6	0.70	0.65	0.03	0.03	0.62	1.52	0.03	0.01	5¼	0.90	0.01	0.01	8.77	1.19	0.52	0.03	0.14	0.11	79	
	991	3 11	1	1¼	46.00	45.48	257	.... 52.5	2.40	2.23	0.08	0.08	2.15	2.91	0.13	0.06	"	1.74	0.05	0.02	16.77	4.39	1.91	0.12	0.38	0.26	68	
	992	3 04	1	1½	52.50	51.48	334	.... 66.5	4.16	3.87	0.11	0.11	3.76	3.78	0.22	0.10	"	2.27	0.08	0.04	21.80	7.40	3.22	0.17	0.75	0.58	77	
	993	3 02	1	1¾	23.91	23.39	452	.... 54.6	7.63	7.10	0.17	0.16	6.94	5.12	0.41	0.18	"	3.07	0.15	0.07	29.52	13.57	5.88	0.25	1.40	1.15	82	
	994	3 02	1	2	17.80	17.30	517	.... 60.7	10.35	9.63	0.23	0.22	9.41	5.86	0.53	0.23	"	3.52	0.19	0.08	33.79	17.74	7.69	0.37	2.17	1.80	83	
	995	3 14	2	1¼	41.84	41.32	300	.... 42.7	0.95	0.88	0.08	0.08	0.80	3.19	0.18	0.08	"	1.91	0.06	0.03	9.79	1.49	0.65	0.13	0.31	0.18	58	
	996	3 01	2	1¾	30.34	29.84	509	.... 39.0	2.70	2.51	0.22	0.21	2.30	5.78	0.52	0.22	"	3.46	0.19	0.08	16.61	4.29	1.87	0.35	0.86	0.51	59	
	997	3 05	2	1¾	44.51	44.01	619	.... 59.5	4.01	3.73	0.30	0.28	3.45	7.03	0.77	0.33	"	4.21	0.28	0.12	20.20	6.33	2.77	0.49	1.29	0.80	62	
	998	3 02½	2	2	33.98	33.48	720	.... 54.8	5.45	5.05	0.41	0.38	4.67	8.17	1.04	0.45	"	4.90	0.37	0.16	23.50	8.58	3.73	0.67	1.77	1.10	62	
	999	3 04½	2	2¼	17.34	16.84	848	.... 48.2	7.51	6.99	0.53	0.50	6.49	9.64	1.44	0.62	"	5.78	0.52	0.22	27.68	11.93	5.18	0.90	2.43	1.53	63	
Outlet No. 1.	1022	3 15	1	¾	67.38	66.86	135	.... 67.7	0.80	0.73	0.02	0.02	0.71	1.52	0.03	0.01	"	0.90	0.01	0.01	8.83	1.20	0.53	0.02	0.21	0.19	90	
	1023	3 10	1	1¼	46.06	45.54	257	.... 53.5	2.53	2.37	0.07	0.07	2.30	2.91	0.13	0.06	"	1.74	0.05	0.02	16.77	4.39	1.91	0.11	0.52	0.41	79	
	1024	3 04	1	1½	52.86	52.34	338	.... 68.0	4.38	4.07	0.10	0.10	3.97	3.83	0.23	0.10	"	2.30	0.08	0.04	22.05	7.57	3.28	0.16	0.89	0.73	82	
	1025	3 01½	1	1¾	24.31	23.81	456	.... 55.7	8.30	7.72	0.16	0.15	7.57	5.16	0.42	0.18	"	3.09	0.15	0.07	29.79	13.80	5.99	0.26	1.91	1.65	86	
	1026	3 07	1	2	18.08	17.58	522	.... 62.7	11.14	10.35	0.24	0.23	10.12	5.93	0.55	0.24	"	3.55	0.20	0.09	34.10	18.09	7.85	0.38	2.74	2.36	86	

Zero corr. = 0.50 and 0.52.

EXPERIMENTS ON CHAPMAN THREE-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 52°. Temperature of Atmosphere, 75°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)−(9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fricт'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd Bbl. (9)−[(15)−(12)].	Act'l Fricт'n Losses. From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)−[(17)−(12)].	Act'l Fricт'n Losses. From Top of Hyd. Bbl. to 2½″ Piez'r. (19)−(18).	Loss at Nozzle in Per Cent. of Total Loss. (20)×100÷(19).	Remarks.
	June 16. 1898. m. s.		in.	g'ge	cor.		g'e.	cor.	g'ge	corr.	g'e	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
1160	4 00	1	¾	65.24	64.70	133	....	66	0.97	0.92	0.06	0.06	0.86	1.51	0.03	0.01		1.15	0.02	0.01	8.69	1.17	0.52	0.06	0.41	0.35	85	Repeated.
1161	3 00	1	1¼	50.80	50.26	266	..	59	4.14	3.87	0.43	0.40	3.47	3.00	0.14	0.06		2.28	0.08	0.03	17.38	4.70	2.04	0.43	1.89	1.46	77	
1162	3 00	1	1½	45.63	45.09	252	...	53	3.50	3.26	0.17	0.16	3.10	2.85	0.12	0.05		2.17	0.07	0.03	16.44	4.21	1.83	0.18	1.48	1.30	88	
1163	5 00	1	¾	67.29	66.75	135	....	67.6	1.00	0.95	0.06	0.06	0.89	1.52	0.03	0.01		1.16	0.02	0.01	8.83	1.20	0.52	0.06	0.44	0.38	86	
1164	2 00	1	1¼	50.24	49.70	328	....	63.3	5.60	5.21	0.21	0.20	5.01	3.72	0.21	0.09		2.84	0.13	0.06	21.41	7.13	3.11	0.23	2.19	1.96	89	
1165	3 00	1	1½	24.10	23.58	454	....	54.0	11.02	10.26	0.49	0.45	9.81	5.16	0.41	0.18		3.93	0.24	0.10	29.66	13.68	5.94	0.53	4.50	3.97	88	
1166	3 00	1	2	17.99	17.47	520	....	60.1	15.60	13.96	0.63	0.50	13.46	5.91	0.54	0.23		4.50	0.31	0.13	33.97	17.94	7.78	0.60	6.41	5.81	91	
1167	5 00	2	1¼	36.44	35.90	280	....	40.1	1.28	1.20	0.32	0.30	1.00	3.17	0.15	0.07		2.41	0.09	0.04	9.16	1.29	0.57	0.23	0.70	0.47	67	Tested with two 2½″ piezometers.
1168	3 00	2	1½	30.46	29.94	511	....	39.2	3.96	3.67	0.62	0.57	3.10	5.80	0.52	0.22		4.42	0.30	0.13	16.68	4.33	1.88	0.66	2.01	1.35	67	
1169	3 00	2	1¾	41.84	41.32	600	...	55.7	5.36	5.00	0.89	0.81	4.19	6.81	0.72	0.31		5.19	0.42	0.18	19.60	5.94	2.58	0.94	2.73	1.79	66	
1170	3 00	2	2	33.51	32.99	714	....	54.8	7.63	7.10	1.25	1.16	5.94	8.12	1.02	0.44		6.18	0.59	0.25	23.29	8.43	3.12	1.35	4.42	3.07	69	
1171	3 00	2	2½	17.24	16.72	844	....	48.1	10.90	10.11	1.81	1.67	8.44	9.59	1.43	0.62		7.14	0.79	0.34	27.53	11.82	3.67	1.95	7.06	5.11	72	

Corresponds to diameter of 6¼″.

Zero corr. = 0.52 and 0.54.

EXPERIMENTS ON CHAPMAN THREE-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 52°. Temperature of Atmosphere, 75°.

1	2	3	4	5		6	7	8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)	Measured Loss between 6″ Piezom'r and 2½″ Piezom'r. (Lbs.)		Measured Loss between 6″ Piezom'r and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)–(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actʼl Frictʼn Losses. From 6″ Piezʼr to Gauge Connectʼn at Top of Hyd. Bbl. (9)–[(15)–(12)].	Actʼl Frictʼn Losses. From 6″ to 2½″ Piezʼrs Equals the Total Loss. [(17)–(18)].	Actʼl Frictʼn Losses. From Top of Hyd. Bbl. to 2½″ Piezʼr. (19)–(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	June 16, 1898.																										
	m.	s.	in.	ga'ge.	corr.		g'e.	cor.	ga'ge.	cor.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
1172	3	00	3	1½	19.41	18.91	406	....	20.5	1.36	1.25	0.38	0.36	0.89	4.60	0.33	0.15		3.50	0.19	0.08	8.83	1.20	0.53	0.43	0.87	0.44	51	
1173	2	00	3	1½	40.43	38.93	590	....	46.5	2.88	2.68	0.82	0.76	1.92	6.70	0.70	0.30		5.10	0.40	0.17	12.84	2.55	1.12	0.89	1.86	0.97	52	
1174	3	00	3	1½	19.47	18.97	407	....	21.	1.37	1.26	0.38	0.35	0.90	4.61	0.33	0.15		3.51	0.19	0.08	8.87	1.21	0.53	0.43	0 88	0.45	51	
1175	3	00	3	1½	40.64	40.14	590	....	46.9	2.82	2.63	0.79	0.73	1.90	6.70	0.70	0.30		5.11	0.40	0.17	12.84	2.55	1.12	0.86	1.81	0.95	52	
1176	3	00	3	2	37.03	36.53	750	....	48.7	4.62	4.39	1.24	1.15	3.15	8.52	1.13	0.49		6.49	0.65	0.28	16.32	4.15	1.80	1.36	2.99	1.63	55	
1177	3	00	3	2½	20.38	19.88	921	....	39.9	7.11	6.61	1.94	1.79	4.82	10.46	1.69	0.73		7.97	0.99	0.43	20.03	6.23	2.71	2.09	4.63	2.54	55	
1178	3	30	Outlets Nos. 2 & 3	1½	42.15	41.62	302	....	43.1	1.31	1.29	0.24	0.23	0 97	3.42	0.18	0.08	Corresponds to diameter of 6″.	2.07	0.07	0.03	9.88	1.50	0.66	0.28	0.62	0.34	55	
1179	3	00	2	1½	31.18	30.68	517	....	41.2	3.98	3.70	0.62	0.58	3.12	5.87	0.53	0.23		4.47	0.31	0.14	16.88	4.44	1.93	0.67	2.00	1.33	66	
1180	3	00	2	1½	44.71	44.21	620	....	60.9	5.70	5.30	0.91	0.84	4.46	7.04	0.77	0.33		5.36	0.45	0.19	20.23	6.37	2.77	0.98	2.86	1.88	66	*
1181	3	00	2	2	32.83	32.33	707	....	54.8	7.54	7.01	1.20	1.11	5.90	8.03	1.00	0.43		6.12	0.58	0.25	23.06	8.27	3.58	1.29	3.86	2.57	67	
1182	3	00	2	2½	16.23	15.73	819	....	49.9	10.60	9.89	1.62	1.50	8.49	9.31	1.35	0.59		7.09	0.78	0.34	26.73	11.14	4.83	1.75	5.65	3.90	69	

Zero corr. 0.50.
" " 0.53.

* Tested with one 2½″ piezometer attached to outlet No. 3.

EXPERIMENTS ON CHAPMAN THREE-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 52°. Temperature of Atmosphere, 75°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Teste.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)−(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hyd't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresp'd'g to Column 16.		Act'l Frict'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)−[(15)−(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)−[(17)−(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)−(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	June 16, 1898.																											
	m. s.		in.	g'ge.	corr.		g'e.	cor.	g'ge.	corr.	g'e	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
1183	4 00	1	1	65.53	65.00	134	....	66.1	0.97	0.92	0.05	0.05	0.87	1 52	0.03	0.01	=	1.16	0.02	0.01	8.76	1.19	0.52	0.05	0.41	0.36	88	
1184	3 00	1	1	44 47	45.94	254	....	54.7	3.45	3.19	0.17	0.16	3.03	2.88	0.13	0.06		2.19	0.07	0.03	16.57	4.28	1.86	0.19	1.39	1.20	86	
1185	3 00	1	1	44.03	45.50	253	....	55.0	3.51	3.29	0.18	0.18	3.11	2.87	0.13	0.06		2.19	0.07	0.03	16.50	4.24	1.84	0.21	1.51	1.30	86	
1186	3 00	1	1	40.47	48.94	326	....	64.4	5.52	5.13	0.29	0.28	4.85	3.69	0.21	0.09		2.81	0.11	0.05	21.29	7.04	3.05	0.32	2.15	1.83	85	
1187	3 00	1	1	25.26	24.76	465	....	59.4	11.40	10.59	0.59	0.54	10.05	5.27	0.44	0.19		4.01	0.25	0.11	30.38	14.34	6.22	0.62	4.56	3.94	86	
1188	3 00	1	2	17.46	16.96	512	....	61.5	14.14	13.17	0.74	0.68	12.49	5.82	0.53	0.23	○ > 6⅜″	4.43	0.31	0.14	33.47	17.40	7.55	0.77	5.85	5.08	87	
	June 21, 1898.																Area of Hexagon area cir. 6¼ diam.											
1192	3 00	Outlets	1½	42.21	41.71	302	....	46.0	1.40	1.29	0.21	0.20	1.09	3.42	0.18	0.08		2.07	0.07	0.03	9.88	1.50	0.66	0.25	0.71	0.46	65	
1193	3 00	No. 2 & 3.	1⅝	30.74	30.27	514	....	42.1	4.07	3.80	0.64	0.59	3.21	5.84	0.53	0.23		4.45	0.31	0.14	16.78	4.39	1.91	0.68	2.12	1.44	68	
1194	3 00	2 2 2 2 2	1¾	42.38	41.91	603	....	59.0	5.48	5.08	0.82	0.76	4.32	6.86	0.73	0.32		5.22	0.42	0.18	19.70	6.01	2.62	0.80	2.78	1.98	71	
1195	3 00	2	2	34.37	33.90	723	....	59.0	7.94	7.37	1.19	1.10	6.27	8.21	1.62	0.70		6.25	0.61	0.26	23.60	8.65	3.76	1.54	4.31	2.77	64	
1196	3 00	2	2½	17.50	17.03	856	....	52.9	11.01	10.24	1.72	1.59	8.65	9.72	1.95	0.84		7.40	0.85	0.37	27.94	12.17	5.28	2.05	5.80	3.74	64	
	June 22.																											
1229	3 00	2	2½	17.15	16.65	844	....	50.6	10.83	10.06	1.71	1.59	8.47	9.59	1.43	0.62		7.14	0.79	0.34	27.53	11.82	3.67	1.87	7.01	5.14	73	

* Repeat. † Tested with two 2½-inch piezometers.

Zero corr. = 0.50 } Nos. 1183–1188 inclusive.
" " 0.53 }
" " 0.47 } " 1192–1196 "
" " 0.50 }
" " 0.50 No. 1229.

EXPERIMENTS ON CHAPMAN FOUR-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temper..ture of Water, 38°. Temperature of Atmosphere, 14°–34°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diame'er of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entr'ce. (Lbs.)		Measured Loss between 6″ Piezometer and 2″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2″ Piezometer. (8)—(9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fricт'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hydrant Bbl. (9)—(15)—(12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Jan. 4–5.																											
	m. s.			ga'ge.	cor.		g'e.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
833	5 31	4	2	27.3	26.88	1075	44.2	47.5	5 91	5.50	0.4	0.37	5.13	12.20	2.31	1.00	8¼	6.45	0.65	0.28	17.57	4.80	2.08	1.09	4.42	3.33	75	
834	5 43½	4	2	19.55	19.13	902	31.0	34.3	4.1	3.83	0.29	0.27	3.56	10.23	1.63	.71	"	5.41	0.46	0.20	14.68	3.36	1.47	0.78	3.07	2.29	75	
835	6 43	4	2	20.2	19.78	553	22.0	25.3	1.5	1.40	0.1	0.10	1.30	6.27	0.61	0.26	"	3.32	0.17	0.07	9.02	1.26	0.56	0.29	1.10	0.81	74	
836	6 00½	4	2	35.25	35.83	742	42.0	45.3	2.8	2.60	0.2	0.19	2.41	8.42	1.10	0.48	"	4.45	0.31	0.13	12.08	2.37	0.99	0.54	2.09	1.55	74	
837	5 57½	3	2	35.55	36.13	748	46.4	49.7	3.83	3.56	0.21	0.19	3.37	8.51	1.12	0.49	"	4.50	0.31	0.13	16.25	4.12	1.79	0.55	2.26	1.71	76	
838	5 43½	3	2½	19.42	19.00	900	37.4	40.7	5.6	5.22	0.31	0.29	4.93	10.21	1.62	0.70	"	5.40	0.46	0.20	19.61	5.98	2.60	0.79	3.32	2.53	76	
839	6 13	3	1¾	41.03	40.61	596	45.0	48.3	2.37	2.19	0.13	0.12	2.07	6.77	0.71	0.31	"	3.58	0.20	0.09	13.00	2.62	1.14	0.34	1.36	1.02	75	
840	5 28½	3	1½	19.04	18.62	403	30.0	33.3	1.15	1.06	0.10	0.10	0.96	4.56	0.33	0.14	"	2.41	0.09	0.04	8.77	1.18	0.52	0.20	0.68	0.48	71	

Zero corr. = 0.42.

EXPERIMENTS ON CHAPMAN FOUR-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 34°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fric't'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—[(15)—(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—[(17)—(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss (20)×100÷(19).	Remarks.
	Jan. 5. m. s.		in.	g'ge.	cor.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
841	7 31	2	1¼	42.21	41.76	302	41.6	44.9	1.25	1.16	.067	0.07	1.09	3.42	0.18	0.08	8½	1.81	0.05	0.02	9.87	1.50	0.66	0.13	0.58	0.45	78	
842	5 49½	2	1¼	30.33	29.91	510	38.0	41.3	3.55	3.30	.11	0.10	3.20	5.79	0.52	0.22	" "	3.06	0.15	0.07	16.63	4.32	1.88	0.25	1.64	1.39	85	
843	7 06½	2	1¼	44.88	44.46	622	58.0	61.3	5.27	4.93	.20	0.19	4.74	7.05	0.77	0.33	" "	3.73	0.22	0.10	20.30	6.40	2.78	0.42	2.48	2.06	83	
844	5 07	2	2	33.92	33.50	719	53.3	56.6	7.12	6.62	.30	0.28	6.34	8.17	1.03	0.45	" "	4.32	0.29	0.13	23.46	8.56	3.72	0.60	3.35	2.75	82	
845	6 22½	1	1¼	24.44	24.02	458	57.0	60.3	11.0	10.22	.10	0.10	10.12	5.19	0.42	0.18	" "	2.75	0.12	0.05	29.92	13.92	6.03	0.23	4.37	4.14	95	
846	6 38½	1	1¼	52.77	52.32	337	65.4	68.7	5.81	5.40	.07	0.07	5.33	3.82	0.23	0.10	" "	2.02	0.06	0.03	22.00	7.52	3.27	0.14	2.23	2.09	94	
847	5 44½	1	1¼	46.45	46.00	254	51.8	55.1	3.43	3.18	.03	0.03	3.15	2.88	0.13	0.06	" "	1.52	0.04	0.02	16.58	4.28	1.87	0.07	1.37	1.30	95	
848	5 32	1	¾	66.45	66.00	134	64.2	67.5	0.97	0.90	.01	0.01	0.89	1.52	0.03	0.01	" "	0.80	0.01	....	8.77	1.18	0.51	0.02	0.40	0.38	95	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON COFFIN GATE HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 44°. Temperature of Atmosphere, 47°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Col. umn 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Frict'n Losses. From 6″ Piez'r to Gauge Connection at Top of Hydrant Bbl. (9)—(15)—(12).	Act'l Frict'n Losses. From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12).	Act'l Frict'n Losses. From Top of Hyd. Bbl. to 2½″ Piez'r. (19)–(18).	Losses at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Nov. 10, 1897.		in.	g'e.	cor.	in.	g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
24	10 min.	1	¾	72.4	71.0	140	71.2	74.5	0.88	0.82	0.15	0.15	0.67	1.58	0.04	....	6	1.58	0.04	....	9.17	1.29	0.57	0.15	0.25	0.10	40	
25	"	1	1⅛	46.6	45.2	252	52.8	56.1	2.90	2.70	0.35	0.33	2.37	2.83	0.12	0.04	"	2.83	0.12	0.04	16.47	4.20	1.83	0.33	0.91	0.58	64	
26	"	1	1½	52.8	51.4	334	66.1	69.4	4.83	4.50	0.60	0.56	3.94	3.78	0.29	0.13	"	3.78	0.29	0.13	21.87	7.40	3.23	0.56	1.40	0.84	60	
27	"	1	1¾	24.2	23.0	448	57.7	61.0	9.20	8.56	1.20	1.10	7.46	5.06	0.41	0.18	"	5.06	0.41	0.18	29.28	13.26	5.76	1.10	2.98	1.88	63	
28	"	1	2	17.5	16.2	502	61.5	64.8	12.34	11.44	1.60	1.48	9.96	5.70	0.51	0.22	"	5.70	0.51	0.22	32.81	16.70	7.26	1.48	4.40	2.92	66	
29	"	2	1½	65.4	64.0	373	66.7	70.0	2.18	2.02	0.75	0.70	1.32	4.26	0.28	0.12	"	4.26	0.28	0.12	12.20	2.29	1.00	0.70	1.14	0.44	38	
30	"	2	1⅝	31.4	30.1	514	38.9	42.2	4.16	3.85	1.40	1.28	2 57	5.84	0.53	0.23	"	5.84	0.53	0.23	16.82	4.39	1.91	1.28	2.17	0.89	41	
31	"	2	1¾	44.6	43.3	615	58.0	61.3	5.99	5.57	2.10	1.98	3.59	6.99	0.76	0.33	"	6.99	0.76	0.33	20.12	6.29	2.74	1.98	3.16	1.18	37	
32	"	2	2	34.2	32.9	713	53 6	56.9	8.14	7.55	2.76	2.55	5.00	8.10	1.01	0.44	"	8.10	1.01	0.44	23.29	8.42	3.67	2.55	4.32	1.77	41	
33	"	2	2½	18.0	16.7	844	49.5	52.8	11.51	10.67	3.92	3.65	7.02	9.58	1.42	0.62	"	9.58	1.42	0.62	27.52	11.80	5.12	3.65	6.17	2.52	41	

EXPERIMENTS ON COFFIN GATE HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 42°. **Temperature of Atmosphere, 37°.**

[1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (7)-(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fric't'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd Bbl. (9)—[(15)—(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—[(17)—(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
			in.	g'e.	corr.		g'e.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
189	November 15, 1897. 10 minutes.	1	¾	65.8	64.5	133	64.4	67.7	0.80	0.75	0.10	0.10	0.65	1.51	0.03	0.01	6	1.51	0.03	0.01	8.69	1.17	0.51	0.10	0.25	0.15	60	
190		1	1¼	46.6	45.3	255	51.9	55.2	2.90	2.69	0.40	0.37	2.32	2.89	0.13	0.06	"	2.89	0.13	0.06	16.64	4.32	1.88	0.37	0.87	0.50	57	
191		1	1½	53.2	51.9	336	64.9	68.2	5.00	4.66	0.70	0.65	4.01	3.80	0.22	0.10	"	3.80	0.22	0.10	21.96	7.50	3.27	0.65	1.49	0.84	56	
192		1	1¾	24.8	23.5	453	52.0	55.3	9.42	8.78	1.25	1.16	7.62	5.14	0.41	0.18	"	5.14	0.41	0.18	29.59	13.62	5.91	1.16	3.05	1.89	62	
193		1	2	19.1	17.9	526	60.8	64.1	13.30	12.35	1.75	1.63	10.72	5.98	0.55	0.24	"	5.97	0.55	0.24	34.38	18.37	7.98	1.63	4.61	2.98	65	
194		2	1½	65.7	64.4	374	67.0	70.3	2.22	2.07	0.80	0.75	1.32	4.13	0.28	0.12	"	4.13	0.28	0.12	12.22	2.31	1.00	0.75	1.19	0.44	37	
195		2	1¾	44.1	42.9	610	57.6	60.9	5.98	5.55	2.05	1.93	3.62	6.93	0.75	0.32	"	6.93	0.75	0.32	19.92	6.15	2.67	1.93	3.20	1.27	40	
196		2	1¾	30.1	28.9	503	38.0	41.3	4.10	3.82	1.45	1.34	2.48	5.72	0.51	0.22	"	5.72	0.51	0.22	16.41	4.19	1.82	1.34	2.22	0.88	40	
197		2	2	34.9	33.7	720	54.0	57.3	7.99	7.41	2.80	2.60	4.81	8.17	1.04	0.45	"	8.17	1.04	0.45	23.53	8.61	3.74	2.60	4.12	1.52	37	
199		2	2½	18.4	17.16	859	40.9	44.2	12.03	11.18	3.91	3.63	7.55	9.76	1.48	0.64	"	9.76	1.48	0.64	28.03	12.26	5.32	3.63	6.50	2.87	44	

EXPERIMENTS ON COFFIN COMPRESSION HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 44°. Temperature of Atmosphere, 45°–47°.

1	2	3	4	5		6	7	8		9		10	11	12		13	14	15		16	17		18	19	20	21	
								Measured Loss between 6″ Piezometer and															Act'l Frict'n Losses.				
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance (Lbs.)	2½″ Piezometer. (Lbs.)		Top of Hydr't Bbl. (Lbs.)		Loss between Hydr. at Bbl. and 2½″ Piezom'r. (8) – (9).	Velocity at 6″ Piezometer.	Head Correspond'g to Col. 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9) – (15) – (12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8) – (17) – (12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19) – (18).	Loss at Nozzle in Per Cent of Total Loss. (20) × 100 ÷ (19).	Remarks.
			in.	g'e.	cor.		g'e.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
34	Nov. 10, 1897. 10 min.	1	¾	74.7	73.3	142	73.6	76.9	0.84	0.78	0.10	0.10	0.68	1.59	0.04	0.02	6	1.59	0.04	0.02	9.32	1.33	0.58	0.10	0.22	0.12	55	
35	"	1	1¼	46.5	45.1	252	52.6	55.9	2.71	2.50	0.40	0.37	2.13	2.84	0.12	0.05	"	2.84	0.12	0.05	16.44	4.21	1.83	0.37	0.72	0.35	49	
36	"	1	1½	53.1	51.7	335	65.8	69.1	4.40	4.07	0.50	0.47	3.60	3.80	0.22	0.09	"	3.80	0.22	0.09	21.93	7.45	3.24	0.47	0.92	0.45	49	
37	"	1	1¾	24.2	23.0	448	53.7	57.0	8.27	7.67	0.90	0.84	6.83	5.06	0.41	0.18	"	5.06	0.41	0.18	29.25	13.28	5.77	0.84	2.08	1.24	60	
38	"	1	2	19.1	17.9	526	63.2	66.5	11.68	10.83	1.08	1.00	9.83	5.97	0.55	0.23	"	5.97	0.55	0.23	34.38	18.32	7.96	1.00	3.10	2.10	68	
39	Nov. 11, 1897. 10 min.	2	1¼	63.1	61.7	367	65.0	68.3	1.81	1.69	0.45	0.41	1.28	4.15	0.27	0.12	"	4.15	0.27	0.12	11.96	2.21	0.96	0.41	0.85	0.44	52	
40	"	2	1½	29.8	28.6	500	37.4	40.7	3.60	3.32	0.95	0.88	2.44	5.68	0.50	0.22	"	5.68	0.50	0.22	16.37	4.14	1.80	0.88	1.74	0.86	49	
41	"	2	1¾	44.0	42.7	610	57.2	60.5	5.32	4.95	1.40	1.29	3.66	6.93	0.75	0.33	"	6.93	0.75	0.33	19.95	6.16	2.68	1.29	2.60	1.31	50	
42	"	3	2	33.0	31.8	702	52.2	55.5	6.93	6.44	1.86	1.71	4.73	7.96	0.99	0.43	"	7.96	0.99	0.43	22.93	8.16	3.54	1.71	3.33	1.62	49	
43	"	2	2½	17.2	15.9	828	46.4	49.7	9.84	9.13	2.70	2.52	6.61	9.41	1.37	0.60	"	9.41	1.37	0.60	27.08	11.37	4 93	2.52	4.80	2.28	47	

EXPERIMENTS ON COFFIN COMPRESSION HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 40°. Temperature of Atmosphere, 40°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezom't'r and 2½″ Piezom't'r (Lbs.)		Measured Loss between 6″ Piezom't'r and Top of Hydrant Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydrant Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Frict'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—[(15)—(12)].	Act'l Frict'n Losses. From 6″ to 2½″ Piez'rs Equals the Total Loss. 8—[(17)—(12)].	Act'l Frict'n Losses. From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Nov. 22, 1897.		in.	g'e.	cor.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
265	10 min.	1	¾	66.3	65.0	134	63.7	67.0	0.76	0.70	0.10	0.10	0.60	1.52	0.03	0.02	6	1.52	0.03	0.02	8.77	1.18	0.51	0.10	0.21	0.11	52	
266	"	1	1¼	46.4	45.1	251	51.2	54.5	2.67	2.50	0.20	0.20	2.30	2.84	0.12	0.05	"	2.84	0.12	0.05	16.38	4.20	1.83	0.20	0.72	0.50	69	
267	"	1	1½	53.1	51.8	335	63.9	67.2	4.50	4.20	0.40	0.37	3.83	3.80	0.22	0.10	"	3.80	0.22	0.10	21.87	7.43	3.24	0.37	1.06	0.69	65	
268	"	1	1¾	23.9	22.7	445	48.0	51.3	8.26	7.68	0.75	0.70	6.98	5.03	0.40	0.17	"	5.03	0.40	0.17	29.05	13.15	5.70	0.70	2.15	1.45	67	
269	"	1	2	18.7	17.5	520	54.8	58.1	11.60	10.80	1.05	0.98	9.82	5.90	0.54	0.23	"	5.90	0.54	0.23	33.98	17.95	7.78	0.98	3.25	2.27	70	
270	"	2	1¼	66.4	65.1	376	67.4	70.7	1.90	1.75	0.50	0.47	1.28	4.26	0.28	0.12	"	4.26	0.28	0.12	12.28	2.32	1.01	0.47	0.86	0.39	45	
271	"	2	1½	30.5	29.3	506	38.5	41.8	3.60	3.35	0.95	0.87	2.48	5.74	0.51	0.22	"	5.74	0.51	0.22	16.50	4.24	1.84	0.87	1.73	0.86	50	
272	"	2	1¾	45.3	44.1	620	57.6	60.9	5.38	5.00	1.47	1.33	3.65	7.04	0.77	8.33	"	7.04	0.77	0.33	20.23	6.37	2.77	1.35	2.56	1.21	47	
273	"	2	2	34.3	33.1	718	53.6	56.9	7.30	6.80	1.95	1.80	5.00	8.15	1.03	0.44	"	8.15	1.03	0.44	23.40	8.52	3.70	1.80	3.54	1.74	49	
274	"	2	2½	17.7	16.5	840	48.9	52.2	10.40	9.70	2.75	2.55	7.15	9.55	1.41	0.61	"	9.55	1.41	0.61	27.45	11.71	5.08	2.55	5.23	2.68	51	

EXPERIMENTS ON COREY 4-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 42°. **Temperature of Atmosphere, 40°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17	18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. Lbs.		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)–(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hyd't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.	Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)–[(15)–(12)].	From 6″ to 2½″ Piez'r Equals the Total Loss. (8)–(17)–(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)–(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Nov. 15, 1897.		in.	g'e.	cor.		g'e	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	feet.	lbs.	lbs.	lbs.		
117	10 min.	1	¾	71.1	69.8	138	69.2	72.5	0.85	0.70	0.10	.....	0.60	1.52	0.03	0.01	5¼	1.64	0.04	0.02	8.76	1.19	0.52	0.09	0.19	0.10	53	
118	"	1	⅞	66.5	65.2	134	65.1	68.4	0.75	0.70	0.10	0.10	0.60	2.83	0.11	0.05	"	3.09	0.15	0.07	16.44	4.20	1.83	0.36	0.90	0.54	60	
119	"	1	1¼	46.5	45.2	252	52.0	55.3	2.89	2.68	0.40	0.38	2.30	2.83	0.11	0.05	"	4.16	0.27	0.12	22.00	7.52	3.27	0.53	1.36	0.82	60	
120	"	1	1½	53.6	52.3	337	65.1	68.4	4.89	4.53	0.60	0.56	3.97	3.82	0.23	0.10	"	5.51	0.47	0.20	29.25	13.28	5.77	0.82	2.64	1.82	69	
121	"	1	1¾	24.2	23.0	448	51.0	54.3	8.89	8.23	0.90	0.84	7.39	5.06	0.41	0.18	"	6.58	0.67	0.29	34.83	18.86	8.18	1.41	4.07	2.66	65	
122	"	1	2	19.6	18.4	533	60.1	63.4	12.94	12.00	1.55	1.45	10.55	6.04	0.57	0.25	"	6.58	0.67	0.29	34.83	18.86	8.18	1.41	4.07	2.66	65	
123	"	2	1¼	64.6	63.3	371	66.2	69.5	1.90	1.76	0.80	0.75	1.01	4.20	0.27	0.12	"	4.57	0.32	0.14	12.12	2.28	1.00	0.73	0.88	0.15	17	
124	"	2	1½	30.5	29.3	506	37.1	40.4	3.56	3.31	1.30	1.20	2.11	5.74	0.51	0.22	"	6.25	0.61	0.26	16.50	4.24	1.84	1.16	1.69	0.53	31	
125	"	2	1¾	45.6	44.4	621	57.6	60.9	5.36	5.00	1.90	1.76	3.24	7.05	0.77	0.33	"	7.68	0.92	0.40	20.27	6.38	2.78	1.69	2.55	0.86	34	
126	"	2	2	34.6	33.4	720	53.0	56.3	7.24	6.74	2.50	2.34	4.40	8.17	1.04	0.45	"	8.90	1.23	0.53	23.53	8.61	3.74	2.26	3.45	1.19	34	
127	"	2	2¼	18.5	17.3	860	47.0	50.3	10.43	9.70	3.40	3.17	6.53	9.77	1.48	0.64	"	10.63	1.76	0.76	28.07	12.28	5.33	3.05	5.01	1.96	39	

Zero corr. = 1.3 and 1.2.

EXPERIMENTS ON COREY 4-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 40°. Temperature of Atmosphere, 33°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14		15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entr'ce. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection	Velocity in Hydr't Bbl. at Gauge Connection.		Head Corresp'd'g to Column 15.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fricт'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hydrant Bbl. (9) – (15)—(12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12)	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)–(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × (100)÷(19).	Remarks.
			in.	ga'ge.	cor.		g'e.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
237	Nov. 20–22, '97; 10 minutes.	1	1⅛	66.9	65.6	134	65.0	64.3	0.80	0.75	0.13	0.13	0.62	1.51	0.03	0.01	5¼	1.64	0.04	0.02	8.77	1.18	0.52	0.12	0.24	0.12	50	
238		1	1¼	45.8	44.5	250	51.1	50.4	2.70	2.52	0.40	0.37	2.15	2.84	0.12	0.05		3.09	0.15	0.07	16.30	4.14	1.80	0.35	0.57	0.42	55	
240		1	1½	52.8	51.5	345	65.0	63.3	4.50	4.18	0.65	0.62	3.56	3.79	0.22	0.10		4.13	0.27	0.12	21.88	7.43	3.24	0.60	1.04	0.44	42	
241		1	2	24.5	23.3	452	52.6	55.9	8.57	7.98	1.20	1.12	6.86	5.12	0.41	0.18		5.57	0.48	0.21	29.53	13.57	5.88	1.09	2.98	1.19	52	
242		2	2	18.8	17.6	522	60.5	63.8	12.13	11.26	1.6	1.48	9.78	5.92	0.55	0.24		6.44	0.65	0.28	34.12	18.10	7.86	1.44	3.64	2.20	60	
233		2	2	17.6	16.4	837	47.4	50.7	9.69	9.00	2.46	3.22	5.78	9.50	1.40	0.61		10.33	1.66	0.72	27.33	11.64	5.05	3.11	4.56	1.45	32	
234		2	1¾	34.4	33.2	717	53.6	56.9	7.12	6.60	2.60	2.44	4.16	8.05	1.03	0.45		8.76	1.19	0.52	23.39	8.50	3.70	2.37	3.35	0.98	29	
235		2	1½	45.3	44.1	620	58.0	61.3	5.36	5.00	2.00	1.88	3.12	7.04	0.77	0.33		7.66	0.91	0.39	20.24	6.37	2.77	1.82	2.56	0.74	29	
236		2	1¼	30.1	28.9	503	36.4	39.7	3.56	3.33	1.30	1.20	2.13	5.70	0.51	0.22		6.20	0.60	0.26	16.42	4.19	1.82	1.16	1.73	0.57	33	
		2	1	65.3	64.0	373	69.1	69.4	2.00	1.87	0.90	0.83	1.04	4.23	0.28	0.12		4.61	0.33	0.14	12.19	2.30	1.08	0.81	0.99	0.18	18	

EXPERIMENTS ON COREY 4-INCH HYDRANT.

(RUNNING AROUND WORTHINGTON METER.)

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 40°. Temperature of Atmosphere, 33°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)−(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl (9)−[(15)−(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss, (8)−(17)−(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)−(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
			in.	ga'ge.	corr.		ga'ge.	corr.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
255	November 22, 10 minutes.	1	1¼	66.15	64.85	375	67.5	70.80	2.00	1.87	0.80	0.74	1.13	4.24	0.28	0.12	5¼	4.62	0.33	0.14	12.26	2.32	1.02	0.72	0.97	0.25	26	
256		1	1½	30.33	29.13	503	37.18	40.48	3.6	3.36	1.3	1.21	2.15	5.72	0.51	0.22	"	6.20	0.60	0.26	16.42	4.19	1.82	1.17	1.76	0.59	34	
257		1	1¾	45.72	44.52	622	57.9	61.20	5.47	5.07	2.01	1.90	3.17	7.05	0.77	0.33	"	7.68	0.92	0.40	20.31	6.42	2.80	1.83	2.60	0.77	28	
258		1	2	35.03	33.83	722	53.55	56.85	7.27	6.76	2.54	2.38	4.38	8.20	1.04	0.45	"	8.93	1.24	0.54	23.56	8.62	3.75	2.29	3.46	1.17	34	
259		1	2½	21.24	20.04	925	55.0	58.30	11.82	10.99	4.18	3.90	7.09	10.50	1.71	0.74	"	11.43	2.03	0.88	30.22	14.21	6.17	3.76	5.56	1.80	32	
260		1	¾	65.98	64.68	133	64.72	68.02	0.8	0.74	0.1	0.10	0.64	1.52	0.03	0.01	"	1.65	0.04	0.02	8.69	1.17	0.52	0.09	0.23	0.14	61	
261		1	1¼	46.26	44.96	251	51.63	54.93	2.68	2.50	0.4	0.37	2.13	2.83	0.12	0.05	"	3.08	0.15	0.07	16.37	4.20	1.83	0.35	0.72	0.37	51	
262		1	1½	53.17	51.87	335	65.0	68.30	4.59	4.28	0.7	0.64	3.64	3.80	0.22	0.10	"	4.14	0.27	0.12	21.86	7.43	3.24	0.62	1.14	0.52	46	
263		1	1¾	24.38	23.18	450	51.72	55.02	8.59	8.00	1.2	1.11	6.89	5.10	0.41	0.18	"	5.85	0.48	0.20	29.40	13.45	5.84	1.19	2.34	1.15	53	
264		1	2	19.62	18.42	532	62.9	66.20	12.77	11.84	1.75	1.62	10.22	6.03	0.57	0.27	"	6.56	0.67	0.29	34.76	18.78	8.15	1.58	3.94	2.36	60	

Zero corr. = 1.3 and 1.2.

EXPERIMENTS ON COREY 5-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 51°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.).		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actl Frict'n Losses. From 6″ Piez'r to Gauge Connection at Top of Hydrant Bbl. (9)—[(15)—(13)].	Actl Frict'n Losses. From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—[(17)—(12)].	Actl Frict'n Losses. From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Dec. 15, 1897.		in.	g'ge.	cor.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
572	10 min.	1	½	66.37	65.92	134	64.6	67.9	0.7	0.63	0.05	0.05	0.58	1.52	0.03	0.01	6½	1.20	0.02	0.01	8.76	1.19	0.52	0.05	0.12	0.07	58	
573	"	1	1¼	46.16	45.71	253	51.0	54.3	2.48	2.30	0.1	0.10	2.20	2.86	0.13	0.06	"	2.26	0.08	0.04	16.50	4.25	1.84	0.12	0.52	0.40	77	
574	"	1	1½	53.55	53.10	340	65.0	68.3	4.2	3.92	0.2	0.19	3.73	3.85	0.23	0.10	"	3.04	0.14	0.06	22.20	7.66	3.33	0.23	0.69	0.46	67	
575	"	1	1¾	23.78	23.36	451	51.4	54.7	7.84	7.28	0.38	0.36	6.92	5.11	0.41	0.18	"	4.04	0.25	0.11	29.45	13.50	5.86	0.43	1.60	1.17	73	
576	"	1	2	19.61	19.19	545	61.5	64.8	11.66	10.83	0.53	0.50	10.33	6.18	0.59	0.25	"	4.88	0.37	0.16	35.62	19.71	8.55	0.59	2.53	1.94	77	
577	"	2	1¼	65.5	65.05	375	66.1	69.4	1.41	1.29	0.2	0.19	1.10	4.24	0.28	0.12	"	3.35	0.17	0.07	12.22	2.33	1.01	0.24	0.40	0.16	40	
578	"	2	1½	30.36	29.94	510	36.0	39.3	2.65	2.48	0.4	0.37	2.11	5.79	0.52	0.22	"	4.57	0.32	0.14	16.63	4.32	1.88	0.45	0.82	0.37	45	
579	"	2	1¾	45.66	45.24	628	56.1	59.4	4.03	3.75	0.6	0.55	3.20	7.14	0.79	0.34	"	5.64	0.48	0.21	20.50	6.54	2.84	0.68	1.25	0.57	46	
580	"	2	2	34.23	33.81	722	50.9	54.2	5.46	5.06	0.8	0.74	4.32	8.19	1.04	0.45	"	6.47	0.65	0.28	23.57	8.64	3.75	0.91	1.76	0.85	48	
581	"	2	2½	17.54	17.12	855	43.1	46.4	7.83	7.26	1.1	1.02	6.24	9.72	1.46	0.63	"	7.68	0.92	0.40	27.92	12.14	5.27	1.25	2.62	1.37	52	

Zero corr. =0.42 and 0.45.

EXPERIMENTS ON COREY 5-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 39°. **Temperature of Atmosphere, 50°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Frict'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—[(15)—(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Dec. 16.																											
	m. s.		in.	g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
587	5 27⅝	1	¾	66.54	66.09	134	64.9	68.2	0.7	**0.64**	0.05	0.05	0.59	1.52	0.03	0.01	6¼	1.20	0.02	0.01	8.76	1.19	0.52	0.05	0.13	0.08	62	
588	5 57⅝	1	1¼	46.39	45.94	254	51.7	55.0	2.47	2.27	0.1	0.10	2.17	2.88	0.13	0.06	"	2.27	0.08	0.04	16.58	4.28	1.87	0.12	0.46	0.34	74	
589	6 53⅜	1	1½	53 3	52.85	338	65.2	68.5	4.2	3.92	0.2	0.19	3.73	3.84	0.23	0.10	"	3.03	0.14	0.06	22.06	7.57	3.29	0.23	0.73	0.50	68	
590	5 05⅝	1	1¾	24.51	24.09	460	52.7	57.0	7.82	7.25	0.39	0.37	6.88	5.22	0.43	0.19	"	4.12	0.26	0.11	30.04	14.04	6.08	0.45	1.36	0.91	67	
591	11 19	1	2	19.27	18.85	540	60.6	63.9	11.23	10.45	0.53	0.50	9.95	6.13	0.58	0.25	"	4.84	0.36	0.16	35.29	19.34	8.40	0.59	2.30	1.71	74	
	Dec. 15.																											
582	10 min.	2	1½	64.75	64.30	374	66.0	69.3	1.45	1.34	0.2	0.19	1.15	4.23	0.28	0.12	"	3.34	0.17	0.07	12.32	2.31	1.01	0.24	0.45	0.21	47	
583	4 "	2	1¾	30.23	29.81	510	36.0	39.3	2.79	2.51	0.4	0.37	2.14	5.79	0.52	0.22	"	4.57	0.32	0.14	16.63	4.32	1.88	0.45	0.85	0.40	47	
584	5 "	2	1¾	45.58	45.16	628	57.5	60.8	4.10	3.83	0.6	0.55	3.28	7.14	0.79	0.34	"	5.64	0.48	0.21	20.50	6.54	2.84	0.68	1.33	0.65	49	
585	5 "	2	2	33.97	33.55	720	50.7	54.0	6.47	5.06	0.8	0.74	4.32	8.17	1.04	0.45	"	6.46	0.65	0.28	23.53	8.61	3.74	0.91	1.77	0.86	49	
586	5 "	2	2½	17.67	17.25	858	44.6	47.9	7.68	7.12	1.1	1.02	6.10	9.76	1.47	0.64	"	7.71	0.92	0.40	28.00	12.22	5.31	1.26	2.45	1.19	49	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON COREY 5-INCH SPECIAL HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 32°.

1	2		3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.		No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connection at Top of Hydrant Bbl. (9)—[(15)—(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—[(17)—(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
	Dec. 23			in.	g'ge.	corr.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
	m.	s.																											
678	7	13⅝	1	1½	5.37	4.95	208	9.0	12.3	1.55	1.44	0.1	0.10	1.34	2.36	0.08	0.04	6¼	1.86	0.05	0.02	13.58	2.86	1.24	0.12	0.24	0.12	50	
679	7	27⅝	1	"	10.57	10.15	298	21.0	24.3	3.5	3.06	0.2	0.19	2.87	3.37	0.18	0.08	"	2.66	0.11	0.05	19.46	5.88	2.55	0.32	0.59	0.37	63	
680	6	2	1	"	15.45	15.03	352	32.2	35.5	4.92	4.56	0.25	0.23	4.33	4.09	0.26	0.11	"	3.23	0.16	0.07	23.62	8.67	3.77	0.27	0.90	0.63	70	
681	5	16⅜	1	"	20.44	20.02	418	42.3	45.6	6.52	6.05	0.3	0.28	5.77	4.74	0.35	0.15	"	3.75	0.22	0.10	27.28	11.60	5.03	0.33	1.17	0.84	72	
682	5	44	1	"	25.72	25.30	468	54.3	57.6	8.25	7.65	0.39	0.36	7.29	5.31	0.44	0.19	"	4.20	0.27	0.12	30.57	14.53	5.32	0.43	1.52	1.09	72	
683	5	44	1	"	30.28	29.86	510	64.0	67.3	9.7	8.98	0.4	0.37	8.61	5.79	0.52	0.22	"	4.57	0.32	0.14	33.33	17.27	7.49	0.45	1.71	1.26	74	

EXPERIMENTS ON GLAMORGAN 4-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 44°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entr'ce. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11		Diameter of Hydrant Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connection Top of Hydrant Bbl (9)—(13)—(12).	From 6″ to 2½″ Piez'r is Equals the Total Loss. (8)—(17)—(12).	From Top. of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100÷(19).	Remarks.
				ga'ge.	cor.		ga'ge.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
	Dec. 9.		in.																									
468	10 min.	1	1 1/16	66.15	65.70	134	.....		0.81	0.74	0.1	0.10	0.64	1.52	0.03	0.02	4¼	2.42	0.09	0.04	8.57	1.18	0.51	0.08	0.25	0.17	68	
469	"	1	1 1/16	46.52	46.10	254	52.0	55.3	3.00	2.81	0.25	0.23	2.58	2.88	0.13	0.06	"	4.59	0.33	0.14	16.58	4.28	1.87	0.15	1.00	0.85	85	
470	"	1	1 1/16	53.52	53.07	342	66.7	70.0	4.99	4.62	0.4	0.38	4.24	3.87	0.23	0.10	"	6.17	0.59	0.25	22.35	7.77	3.38	0.23	1 34	1.11	83	
471	"	1	1 1/16	24.66	24.24	459	58.6	61.9	9.53	8.87	0.68	0.62	8.25	5.21	0.43	0.19	"	8.31	1.07	0.46	30.00	13.98	6.07	0.35	2.99	2.64	88	
472	"	1	2	18.09	17.67	523	60.2	63.5	12.78	11.87	0.97	0.89	10.98	5.94	0.55	0.24	"	9.47	1.39	0.60	34.18	18.16	7.88	0.53	4.23	3.70	87	
	Dec. 9.																											
473	10 min.	2	1¼	65.55	65.10	376	67.09	70.4	1.84	1.70	0.5	0.47	0.23	4.26	0.28	0.12	"	6.79	0.72	0.31	12.28	2.33	1.02	0.28	0.80	0.52	65	
474	"	2	1¼	30.46	30.04	512	38.4	41.7	3.46	3.20	0.8	0.75	2.45	5.80	0.53	0.23	"	9.25	1.33	0.58	16.72	4.35	1.90	0.40	1.53	1.13	74	
475	"	2	1¼	45.06	44.64	623	58.4	61.7	5.11	4.78	1.14	1.05	3.73	7.07	0.78	0.34	"	11.28	1.98	0.86	20.33	6.42	2.80	0.53	2.32	1.79	77	
476	"	2	2	34.59	34.17	726	53.8	57.1	6.97	6.46	1.56	1.45	5.01	8.24	1.05	0.45	"	13.11	2.67	1.16	23.68	8.72	3.80	0.74	3.11	2.37	76	
477	"	2	2½	17.17	16.75	846	48.5	51.8	9.62	8.94	2.15	2.00	6.94	9.60	1.43	0.62	"	15.31	3.64	1.58	27.61	11.88	5.16	1.04	4.40	3.36	76	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON GLAMORGAN 4-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 44°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2¼″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2¼″ Piezom'r. (8)–(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2¼″ Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)–(15)–(12).	From 6″ to 2¼″ Piez'rs Equals the Total Loss. (8)–(17)–(12).	From Top of Hyd. Bbl. to 2¼″ Piez'r. (19)–(18).	Loss at Nozzle in Per Cent. of Total Loss. (30) × 100 ÷ (19).	Remarks.
	Dec. 9.		in.	g'ge.	cor.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
483	10 min.	1	⅞	66.13	65.68	134	65.0	68.3	0.8	0.74	0.1	0.10	0.63	1.52	0.03	0.01	4¼	2.42	0.09	0.04	8.76	1.19	0.52	0.07	0.23	0.16	70	
484	" "	1	1¼	45.99	45.54	252	51.5	54.8	2.84	2.63	0.2	0.19	2.44	2.83	0.12	0.05	"	4.53	0.32	0.14	16.44	4.20	1.83	0.10	0.85	0.75	88	
485	" "	1	1½	53.79	53.34	340	67.4	70.7	4.84	4.50	0.4	0.37	4.13	3.85	0.23	0.10	"	6.14	0.59	0.25	22.20	7.66	3.33	0.22	1.27	1.05	83	
486	" "	1	1⅝	24.27	23.85	455	54.2	57.5	8.88	8.24	0.7	0.65	7.59	5.16	0.42	0.18	"	8.23	1.05	0.45	29.72	13.54	5.88	0.38	2.54	2.16	85	
487	" "	1	2	18.53	18.11	529	63.5	66.8	12.4	11.51	0.9	0.83	10.68	6.00	0.56	0.24	"	9.57	1.42	0.62	34.57	18.57	8.06	0.45	3.69	3.24	88	
478	5 min.	2	1¼	65.6	65.15	376	67.7	71.0	1.9	1.75	0.5	0.47	1.28	4.26	0.28	0.12	"	6.79	0.72	0.31	12.28	2.33	1.01	0.28	0.86	0.58	67	
479	" "	2	1½	30.39	29.97	512	37.8	41.1	3.4	3.14	0.8	0.74	2.40	5.82	0.53	0.23	"	9.25	1.33	0.58	16.71	4.36	1.89	0.39	1.48	1.09	74	
480	" "	2	1¾	45.16	44.74	623	58.5	61.8	5.18	4.80	1.15	1.06	3.74	7.08	0.78	0.34	"	11.28	1.98	0.86	20.37	6.44	2.80	0.54	2.34	1.80	77	
481	" "	2	2	34.42	34.00	723	54.0	57.3	6.94	6.42	1.58	1.46	4.96	8.21	1.05	0.46	"	13.09	2.66	1.15	23.55	8.63	3.75	0.77	3.13	2.36	83	
482	" "	2	2¼	17.31	16.89	850	48.6	51 9	9.77	9.05	2.17	2.01	7.04	9.66	1.45	0.63	"	15.40	3.69	1.60	27.74	12.00	5.21	1.04	4.47	3.43	85	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON GLAMORGAN 4-INCH HYDRANT.

(LEFT-HAND OUTLET.)

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 38°. Temperature of Atmosphere, 22°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozz'le.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6" Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6" Piezom't'r and 2½" Piezom't'r (Lbs.)		Measured Loss between 6" Piezom't'r and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 24" Piezom't'r. (8)—(9)	Velocity at 6" Piezometer.	Head Corresponding to Column 11.		Diameter of Hydrant Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 24" Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6" Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—[(15)—(12)].	From 6" to 24" Piez'rs Equals the Total Loss. (8)—[(17)—(12)].	From Top of Hyd. Bbl. to 24" Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Dec. 29.																											
	m. s		in.	g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
785	5 35½	1	¾	66.15	65.70	134	64.3	67.6	0.8	0.75	0.1	0.10	0.65	1.52	0.03	0.01	4¼	2.42	0.09	0.04	8.76	1.19	0.52	0.07	0.24	0.17	71	
786	5 44½	1	1½	46.55	46.10	254	51.6	54.9	2.86	2.64	0.25	0.23	2.41	2.88	0.13	0.06	"	4.59	0.33	0.14	16.58	4.28	1.87	0.15	0.83	0.68	82	
787	6 35½	1	1½	53.15	52.70	338	65.5	68.8	4.8	4.47	0.4	0.37	4.10	3.84	0.29	0.13	"	6.12	0.58	0.25	22.06	7.57	3.29	0.25	1.31	1.06	81	
788	6 24½	1	1½	24.25	23.83	455	52.7	56.0	9.0	8.37	0.7	0.65	7.72	5.16	0.42	0.18	"	8.23	1.05	0.45	29.72	13.54	5.88	0.38	2.67	2.29	86	
789	5 17½	1	2	19.36	18.94	540	64.4	67.7	13.09	12.13	1.06	0.99	12.10	6.13	0.58	0.25	"	9.78	1.49	0.65	35.29	19.34	8.40	0.59	3.98	3.39	85	
	Dec. 29.																											
790	5 31½	1	¾	66.36	65.91	134	65.0	68.3	0.8	0.75	0.1	0.10	0.65	1.52	0.03	0.01	"	2.42	0.09	0.04	8.76	1.19	0.52	0.07	0.24	0.17	71	
791	5 43½	1	1½	46.29	45.84	254	51.8	55.1	2.84	2.63	0.25	0.23	2.40	2.88	0.13	0.06	"	4.59	0.33	0.14	16.58	4.28	1.87	0.15	0.82	0.67	82	
792	6 35½	1	1½	53.25	52.80	338	65.4	68.7	4.8	4.47	0.4	0.37	4.10	3.84	0.29	0.13	"	6.12	0.58	0.25	22.06	7.57	3.29	0.25	1.31	1.06	81	
793	6 22½	1	1½	24.32	23.90	457	52.4	55.7	9.07	8.43	0.7	0.65	7.78	5.17	0.42	0.18	"	8.25	1.06	0.46	29.86	13.87	6.02	0.37	2.59	2.22	86	
794	5 16½	1	2	19.18	18.76	538	63.0	66.3	13.1	12.16	1.07	1.00	11.16	6.11	0.48	0.25	"	9.74	1.48	0.64	35.15	19.20	8.39	0.61	4.02	3.41	85	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON BEAUMONT 4-INCH HYDRANT.

(RIGHT-HAND OUTLET.)

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 32°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
									Measured Loss between 6″ Piezometer and															Act'l Fric't'n Losses.				
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		2½″ Piezometer. (Lbs.)		Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9)	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		From 6″ Piez'r to Gauge Connect'n at Top of Hyd Bbl. (9)—(15)—(12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Dec. 30.																											
	m. s.		in.	g'age.	corr.		g'e.	cor.	g'age.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
795	5 34½	1	¾	66.07	65.62	134	64.2	67.5	0.8	0.74	0.1	0.10	0.64	1.52	0.03	0.01	4¼	2.42	0.09	0.04	8.77	1.18	0.51	0.07	0.24	0.17		
796	5 45½	1	1⅛	46.02	45.57	253	51.6	54.9	2.95	2.73	0.25	0.23	2.50	2.87	0.13	0.06	"	4.58	0.33	0.14	16.50	4.24	1.84	0.15	0.95	0.80	84	
797	6 31	1	1¼	53.50	53.05	340	66.7	70.0	5.16	4.82	0.5	0.47	4.35	3.85	0.23	0.10	"	6.14	0.59	0.25	22.20	7.66	3.33	0.32	1.59	1.27	80	
798	6 21	1	1⅜	24.46	24.04	458	56.7	60.0	9.7	9.02	0.87	0.81	8.21	5.18	0.42	0.18	"	8.26	1.06	0.46	29.93	13.92	6.03	0.53	3.17	2.64	83	
799	5 30¾	1	2	17.84	17.42	518	62.7	56.0	12.96	12.02	1.1	1.01	11.01	5.88	0.54	0.23	"	9.38	1.37	0.59	33.86	17.82	7.73	0.65	4.52	3.87	86	
	Dec. 30.																											
800	5 31½	1	¾	65.82	65.37	134	64.3	67.6	0.8	0.74	0.1	0.10	0.64	1.52	0.03	0.01	"	2.42	0.09	0.04	8.77	1.18	0.51	0.07	0.24	0.17	71	
801	5 43½	1	1⅛	45.98	45.53	253	52.1	55.4	2.96	2.74	0.25	0.23	2.51	2.87	0.13	0.06	"	4.58	0.33	0.14	16.50	4.24	1.84	0.15	0.96	0.81	84	
802	6 38	1	1¼	53.51	53.06	340	67.0	70.3	5.15	4.81	0.5	0.47	4.34	3.85	0.23	0.10	"	6.14	0.59	0.25	22.20	7.66	3.33	0.32	1.58	1.26	80	
803	6 20½	1	1⅜	24.38	23.96	458	56.0	59.3	9.62	8.95	0.84	0.77	8.18	5.18	0.42	0.18	"	8.26	1.06	0.46	29.93	13.92	6.03	0.49	3.10	2.61	84	
804	5 27⅝	1	2	18.03	17.61	522	63.7	67.0	13.06	12.12	1.1	1.01	11.11	5.92	0.55	0.24	"	9.44	1.39	0.60	34.12	18.10	7.86	0.65	4.50	3.85	86	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON GLAMORGAN (NO. 2) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 53°. Temperature of Atmosphere, 78°.

1	2	3	4	5		6	7	8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)	Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8) − (9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Frict'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hydrant Bbl. (9) − (15) − (12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8) − (17) − (12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19) − (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	June 25, 1898.		in.	g'ge.	corr.		g'e.	cor.	g'ge.	corr.	g'e.	cor	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
1253	3 min.	1	¾	65.75	65.24	134	....	68.1	1.20	1.11	0.03	0.03	1.08	1.52	0.03	0.01	6¼	1.29	0.03	0.01	8.76	1.19	0.52	0.03	0.69	0.57	95	
1254	"	1	1	46.20	45.69	253	....	57.8	4.30	4.00	0.10	0.10	3.90	2.87	0.13	0.06	"	2.44	0.03	0.04	16.50	4.24	1.84	0.12	2.22	2.10	95	
1255	"	No. 1. 1	1¼	45.52	45.01	313	....	63.0	6.37	5.86	0.13	0.12	6.74	3.54	0.19	0.08	"	3.02	0.14	0.06	20.43	6.49	2.83	0.14	3.11	2.97	95	
1256	"	1	1½	23.28	22.79	446	....	59.6	13.11	12.18	0.24	0.23	11.95	5.03	0.40	0.17	"	4.29	0.29	0.13	29.11	13.32	5.74	0.27	6.61	6.34	96	
1257	"	1	2	15.84	15.35	486	....	61.7	15.87	14.74	0.29	0.27	14.47	5.50	0.47	0.20	"	4.69	0.34	0.15	31.78	15.69	5.81	0.32	8.13	7.81	96	
1258	"	2	1¼	43.40	42.89	306	....	47.1	1.61	1.50	0.15	0.13	1.37	3.47	0.19	0.08	"	2.95	0.14	0.06	10.00	1.53	0.68	0.15	0.90	0.75	83	
1259	"	2	1½	31.31	30.82	508	....	44.5	4.65	4.33	0.30	0.28	4.05	5.77	0.52	0.22	"	4.92	0.38	0.16	16.58	4.28	1.86	0.34	2.69	2.35	87	
1260	"	2	1½	39.93	39.44	587	....	57.6	5.97	5.52	0.41	0.38	5.14	6.66	0.69	0.30	"	5.66	0.50	0.22	19.17	5.72	2.49	0.46	3.33	2.87	86	
1261	"	2	2	30.11	29.62	676	....	53.3	7.94	7.37	0.59	0.54	6.83	7.68	0.91	0.39	"	6.54	0.66	0.29	22.05	7.57	2.93	0.64	4.83	4.19	87	
1262	"	2	2½	15.31	14.82	797	....	49.1	11.39	10.57	0.84	0.77	9.80	9.05	1.27	0.55	"	7.71	0.92	0.40	26.00	10.53	4.57	0.92	6.55	5.63	86	
1263	"	1	¾			61.67	134	....	67.6	1.16	1.07	0.03	0.03	1.04	1.52	0.03	0.01	"	1.29	0.03	0.01	18.76	1.19	0.52	0.03	0.56	0.53	95	
1264	"	No. 2. 1	1¼			46.21	254	....	58.1	4.27	3.99	0.10	0.10	3.88	2.88	0.13	0.06	"	2.45	0.09	0.04	16.57	4.28	1.86	0.12	2.19	2.07	95	
1265	"	1	1½			51.35	334	....	70.3	7.19	6.70	0.16	0.14	6.66	3.79	0.22	0.10	"	3.23	0.16	0.07	21.80	7.40	3.23	0.17	3.57	3.40	95	
1266	"	1	1¾			23.92	457	....	62.2	13.67	12.70	0.20	0.24	12.46	5.08	0.42	0.18	"	4.33	0.29	0.13	29.85	13.87	6.02	0.29	6.86	6.57	96	
1267	"	1	2			16.12	499	....	64.0	16.77	15.56	0.33	0.30	15.26	5.67	0.50	0.21	"	4.84	0.36	0.16	32.62	16.56	7.17	0.35	8.60	8.25	96	

Zero corr. = 0.49 and 0.51.

EXPERIMENTS ON HOLYOKE GATE, TWO-WAY (4″ BBL.) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 48°. Temperature of Atmosphere, 68°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14		15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8) – (9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.		Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hydrant Bbl. (9) – [(15) – (12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8) – [(17) – (12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19) – (18).	Loss at Nozzle in Per. Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	June 10, 1898. m. s.		in.	g'ge.	corr.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
979	3 14⅜	1	¾	68.57	68.15	137	....	69.3	0.85	0.77	0.20	0.18	0.69	1.55	0.03	0.01	5⅞	1.80	0.05	0.02	8.96	1.23	0.54	0.17	0.24	0.07	29	
980	3 14⅝	1	1	46.06	45.54	257	....	54.3	3.03	2.84	0.78	0.71	2.13	2.91	0.13	0.06	"	3.39	0.18	0.08	16.77	4.39	1.91	0.69	0.99	0.30	30	
981	3 11⅝	1	1¼	50.37	49.85	329	....	65.2	4.74	4.40	0.99	0.90	3.50	3.72	0.21	0.09	"	4.33	0.29	0.13	21.48	7.18	3.13	0.86	1.36	0.50	37	
982	3 02	1	1½	24.66	24.17	459	....	60	9.53	8.87	1.95	1.80	7.07	5.20	0.43	0.19	"	6.05	0.57	0.25	29.99	13.98	6.07	1.74	2.99	1.25	42	
983	3 09⅞	1	2	16.67	16.18	500	....	60	11.65	10.82	2.36	2.17	8.65	5.68	0.45	0.20	"	6.61	0.68	0.29	32.69	16.58	7.19	2.08	3.83	1.75	46	
984	3 03⅝	2	1½	39.90	39.38	293	....	41	1.55	1.45	0.84	0.77	0.68	3.32	0.17	0.07	"	3.86	0.23	0.10	9.58	1.41	0.62	0.74	0.90	0.16	18	
985	3 05	2	1½	34.89	34.40	548	....	47.8	5.24	4.90	2.60	2.42	2.48	6.32	0.60	0.26	"	7.24	0.81	0.35	17.90	4.98	2.16	2.33	3.00	0.67	22	
986	3 09	2	1¼	43.36	42.87	610	....	59.8	6.52	6.05	3.27	3.02	3.04	6.93	0.75	0.32	"	8.05	1.01	0.44	19.90	6.16	2.68	2.90	3.70	0.80	22	
987	3 04	2	2	31.78	31.29	697	....	54.2	8.55	7.97	4.26	3.98	3.99	7.91	0.97	0.42	"	9.20	1.32	0.57	22.74	8.03	3.51	3.83	4.88	1.05	22	
988	3 03⅝	2	2½	16.79	16.30	834	....	51	12.42	11.52	6.30	5.87	5.65	9.48	1.39	0.60	"	11.03	1.89	0.82	27.21	11.54	5.01	5.65	7.11	1.46	21	

Zero corr. = 0.49 and 0.52.

EXPERIMENTS ON HOLYOKE GATE, TWO-WAY (4″ BBL.) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 50°. Temperature of Atmosphere, 72°.

	1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16		17		18	19	20	21	
	Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance.		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)–(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.		Head Corresponding to Column 16.		Act'l Fricт'n Losses. From 6″ Piez'r to Gauge Connec'n at Top of Hyd., bl. (9)–(15)–(12).	From 6″ to 2½″ Piez'rs Equals the Total Loss, (8)–(17)–(12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)–(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
		June 11, 1898.																											
		m. s.		in.	g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s		ft.	lbs.	lbs.	lbs.	lbs.		
Outlet No. 1.	1027	3 13	1	¾	66.89	66.37	135	....	68	0.85	0.78	0.16	0.15	0.63	1.52	0.03	0.01	$5\frac{5}{16}$	1.77	0.05	0.02	8.83		1.20	0.53	0.14	0.26	0.12	46	
	1028	3 13	1	1	45.41	44.89	253	....	53.5	2.85	2.64	0.66	0.62	2.02	2.85	0.12	0.05	"	3.32	0.17	0.07	16.50		4.23	1.84	0.60	0.85	0.25	30	
	1029	3 03	1	1	52.34	51.82	336	....	68.7	4.94	4.58	1.03	0.97	3.61	3.80	0.29	0.13	"	4.42	0.30	0.13	21.93		7.47	3.26	0.97	1.45	0.48	33	
	1030	3 01	1	1¼	23.75	23.25	450	....	56	9.11	8.49	1.88	1.73	6.76	5.19	0.41	0.18	"	5.93	0.55	0.24	29.40		13.45	5.83	1.67	2.84	1.17	41	
	1031	3 04	1	2	17.05	16.55	505	....	61	12.91	11.16	2.45	2.27	8.89	5.74	0.51	0.22	"	6.68	0.69	0.30	33.02		16.91	7.34	2.19	4.04	1.85	46	
	1032	3 07	2	1½	43.77	43.55	308	....	45.6	1.63	1.51	0.80	0.74	0.77	3.50	0.19	0.08	"	4.07	0.26	0.11	10.07		1.57	0.68	0.71	0.91	0.20	18	
	1033	3 01	2	1¼	30.48	29.98	506	....	40.6	4.59	4.28	2.28	2.11	2.17	5.74	0.51	0.22	"	6.68	0.69	0.30	16.51		4.24	1.84	2.03	2.66	0.63	24	
	1034	3 01	2	1½	42.92	42.42	608	....	59.7	6.42	5.99	3.23	3.00	2.99	6.92	0.74	0.32	"	8.05	1.01	0.44	19.85		6.11	2.66	2.88	3.65	0.77	21	
	1035	3 01	2	2	34.48	33.98	724	....	59.4	9.14	8.50	4.66	4.33	4.17	8.22	1.05	0.45	"	9.56	1.41	0.61	23.62		8.67	3.77	4.17	5.18	1.01	20	
	1036	3 01	2	2½	17.13	16.63	844	....	52.3	12.69	11.78	6.24	5.81	5.97	9.58	1.43	0.62	"	11.15	1.93	0.84	27.54		11.83	5.14	5.59	7.26	1.67	23	
Outlet No. 2.	1037	3 20	1	¾	66.33	65.81	134	....	67.5	0.85	0.78	0.16	0.15	0.63	1.52	0.03	0.01	"	1.77	0.05	0.02	8.77		1.19	0.52	0.14	0.27	0.13	48	
	1038	3 11	1	1	46.97	46.45	256	....	55.4	2.91	2.70	0.61	0.57	2.13	2.89	0.13	0.06	"	3.36	0.18	0.08	16.70		4.35	1.89	0.55	0.87	0.32	36	
	1039	3 03	1	1½	52.66	52.14	337	....	68.5	4.92	4.56	1.02	0.95	3.61	3.82	0.23	0.10	"	4.44	0.31	0.13	22.00		7.52	3.28	0.92	1.38	0.46	33	
	1040	3 09	1	1½	24.51	24.01	458	.	57.5	9.36	8.68	1.95	1.81	6.87	5.19	0.42	0.18	"	6.04	0.57	0.25	29.92		13.92	6.04	1.74	2.82	1.08	38	
	1041	3 06	1	2	18.48	17.98	528	....	63.3	12.71	11.80	2.55	2.37	9.43	6.00	0.56	0.24	"	6.98	0.76	0.33	34.50		18.50	8.02	2.28	4.02	1.74	43	

Zero corr. = 0.50 and 0.52.

EXPERIMENTS ON HOLYOKE GATE (5¼" BBL.) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 42°. Temperature of Atmosphere, 45°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6" Piezometer at Hydrant Entrance. (Lbs.)		Measured Loss between 6" Piezometer and 2½" Piezometer. (Lbs.)		Measured Loss between 6" Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½" Piezometer. (8) – (9).	Velocity at 6" Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½" Piezometer.	Head Corresponding to Column 16.		Act'l Frict'n Losses. From 6" Piez'r to Gauge Connect'n at Top of Hydrant Bbl. (9) – (15) – (12).	From 6" to 2½" Piez'rs Equals the Total Loss. (8) – (17) – (12).	From Top of Hyd. Bbl. to 2½" Piez'r. (19) – (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Nov. 9.		in.	ga'ge.	corr.		ga'ge.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
5	10 min.	1	1 25/32	65.7	64.30	133	72.8	76.1	0.9	0.84	0.05	0.05	0.79	1.51	0.0	0.01	6¼	1.49	0.03	0.01	8.69	1.17	0.51	0.05	0 34	0.29	85	
6	" "	1	1 25/32	45.8	44.40	250	54.0	57.3	3.0	2.80	0.21	0 20	2.60	2.83	0.1[illegible]	0.05	"	2.72	0.12	0.05	16.32	4.16	1.81	0.20	1.04	0 84	81	
7	" "	1	1 25/32	52.8	51.40	335	67.2	70.5	4.9	4.54	0.4	0.38	4.16	3.80	0.2[illegible]	0.10	"	3.65	0.21	0.09	21.87	7.43	3.24	0.39	1.40	1.01	72	
8	" "	1	1 25/32	28.3	27.10	487	63.3	66.6	10.6	9.87	0.82	0.76	9.11	5.52	0.4	0.21	"	5.30	0.44	0.19	31.84	15.74	6.83	0.78	3.25	2.47	76	
9	" "	2	1 25/32	64.6	63.20	370	67.1	70.4	1.9	1.76	0.45	0.42	1.34	4.19	0.27	0.12	"	4.02	0.25	0.11	12.08	2.27	0.98	0.43	0.90	0.47	52	
10	" "	2	1 25/32	30.6	29.40	506	38.1	41	3.5	3.26	0.8	0.74	2.52	5.74	0.51	0.22	"	5.51	0.47	0.20	16.50	4.24	1.84	0.76	1.64	0.88	54	
11	" "	2	2	34.5	33.30	717	54.1	57.	7.3	6.78	1.8	1.67	5.11	8.14	1.03	0.44	"	7.81	0.95	0.41	33.38	8.50	3.70	1.70	3.52	1.82	52	
12	" "	2	2	17.4	16.20	832	48.0	51.3	9.7	9.00	2.59	2.41	6.59	9.45	1.39	0.60	"	9.07	1.28	0.55	27.14	11.50	5.00	2.46	4.60	2.14	47	

Zero corr. = 1.4 and 1.2.

EXPERIMENTS ON HOLYOKE GATE (5¼" BBL.) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 42°. Temperature of Atmosphere, 45°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6" Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6" Piezometer and 2½" Piezometer. (Lbs.)		Measured Loss between 6" Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½" Piezom'r. (8)—(9).	Velocity at 6" Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 6" Piezometer.	Head Corresponding to Column 16.		Act'l Fricт'n Losses. From 6" Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—(15)—(12).	From 6" to 2½" Piez'rs Equals the Total Loss. (8) — (17)—(12).	From Top of Hyd. Bbl. to 2½" Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19.)	Remarks.
	1897. 10 min.		in.	g'e.	cor.		g'ge.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
62	" "	1	¾	65.6	64.2	134	64.10	67.4	0.75	0.70	0.05	0.05	0.65	1.52	0.03	0.01	6¼	1.46	0.03	0.01	8.76	1.18	0.51	0.05	0.20	0.15		
63	" "	1	1⅛	46.1	44.7	252	51.5	54.8	2.80	2.60	0.25	0.23	2.37	2.83	0.12	0.05	" "	2.72	0.12	0.05	16.44	4.20	1.83	0.23	0.82	0.59	72	
64	" "	1	1½	54.6	53.2	342	67.2	70.5	4.80	4.47	0.40	0.37	4.10	3.87	0.23	0.10	" "	3.71	0.21	0.09	22.32	7.74	3.38	0.38	1.19	0.81	68	
65	" "	1	1⅞	23.3	22.1	438	51.3	54.6	8.36	7.76	0.75	0.70	7.06	4.96	0.39	0.17	" "	4.76	0.35	0.15	28.60	12.75	5.53	0.72	2.40	1.68	70	
66	" "	1	2	18.9	17.7	524	61.2	64.5	12.60	11.70	1.03	0.97	10.73	5.95	0.55	0.24	" "	5.71	0.51	0.22	34.23	18.22	7.91	0.99	4.03	3.04	75	
67	" "	2	1½	65.8	64.4	374	67.0	70.3	1.90	1.76	0.43	0.40	1.36	4.23	0.28	0.12	" "	4.06	0.26	0.11	12.32	2.31	1.01	0.41	0.87	0.46	53	
68	" "	2	1¾	29.7	28.5	498	36.1	39.4	3.40	3.16	0.85	0.78	2.38	5.66	0.50	0.22	" "	5.43	0.46	0.20	16.26	4.10	1.78	0.80	1.60	0.80	50	
69	" "	2	1⅞	45.2	44.0	620	57.4	60.7	5.23	4.89	1.29	1.20	3.69	7.04	0.77	0.33	" "	6.76	0.71	0.31	20.23	6.37	2.77	1.32	2.45	1.23	50	
70	" "	2	2	33.8	32.6	710	51.6	54.9	6.96	6.46	1.70	1.57	4.89	8.07	1.01	0.44	" "	7.74	0.93	0.40	23.16	8.33	3.62	1.61	3.28	1.67	51	
71	" "	2	2¼	17.5	16.3	838	44.7	48.0	10.06	9.37	2.45	2.26	7.11	9.53	1.42	0.61	" "	9.15	1.30	0.57	27.36	11.67	5.06	2.30	4.92	2.62	53	

EXPERIMENTS ON HOLYOKE GATE (6¼" BBL.) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 42°. Temperature of Atmosphere, 40°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6" Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6" Piezometer and 2½" Piezometer. (Lbs.)		Measured Loss between 6" Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½" Piezom'r. (8)—(9).	Velocity at 6" Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½" Piezometer.	Head Corresponding to Column 16.		Act'l Fric't'n Losses. From 6" Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—[15)—(12)].	From 6" to 2½" Piez'rs Equals the Total Loss. (8)—(17)—(12)].	From Top of Hyd. Bbl. to 2½" Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20 × 100 ÷ (19).	Remarks.
			in.	g'e.	cor.		g'e.	cor.	g'age.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
128	November 15, 1897. 10 minutes.	1	¾	65.8	64.5	133	63.5	66.8	0.75	0.70	0.05	0.05	0.65	1.52	0.03	0.01	7	1.12	0.02	0.01	8.69	1.18	0.51	0.05	0.20	0.15	75	
129		1	1¼	45.8	44.5	250	51.0	54.3	2.60	2.42	0.17	0.16	2.26	2.84	0.12	0.05		2.09	0.07	0.03	16.31	4.15	1.81	0.18	0.66	0.48	73	
130		1	1½	53.5	52.2	337	64.2	67.5	4.39	4.08	0.30	0.28	3.80	3.82	0.23	0.10		2.81	0.12	0.05	22.00	7.52	3.27	0.33	0.91	0.58	64	
131		1	1¾	24.2	23.0	458	50.1	53.4	8.09	7.52	0.60	0.56	6.96	5.18	0.42	0.18		3.81	0.23	0.10	29.93	13.92	6.03	0.64	1.67	1.03	62	
132		1	2	19.4	18.2	530	61.1	64.4	12.01	11.16	0.90	0.84	10.32	6.01	0.56	0.24		4.42	0.30	0.13	34.63	18.63	8.08	0.95	3.32	2.37	71	
133		2	1¼	67.0	65.7	377	68.1	71.4	1.70	1.57	0.40	0.37	1.20	4.26	0.28	0.12		3.13	0.15	0.07	12.32	2.34	1.01	0.42	0.68	0.26	38	
134		2	1½	30.4	29.2	504	36.6	39.9	3.12	2.90	0.70	0.65	2.25	5.72	0.51	0.22		4.20	0.27	0.12	16.47	4.22	1.83	0.75	1.29	0.54	42	
135		2	1¾	45.5	44.3	620	57.0	60.3	4.77	4.41	1.10	1.02	3.39	7.04	0.77	0.33		5.17	0.42	0.18	20.23	6.35	2.77	1.17	1.97	0.80	41	
136		2	2	33.9	32.7	710	50.3	53.6	6.22	5.79	1.40	1.30	4.49	8.06	1.01	0.44		5.92	0.55	0.24	23.17	8.33	3.62	1.50	2.61	1.11	43	
137		2	2½	19.0	17.8	873	46.3	49.6	9.61	8.95	2.15	2.00	6.95	9.92	1.52	0.66		7.29	0.83	0.36	28.50	12.66	5.48	2.30	4.13	1.83	44	

EXPERIMENTS ON HOLYOKE COMPRESSION (NO. 1) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 42°. Temperature of Atmosphere, 45°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entrance (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8) – (9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydrant Bbl. at Gauge Connection	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9) – [(15) – (12)].	From 6″ to 2½″ Piez'r Equals the Total Loss. (8) – [(17) – (12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19) – (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Nov. 9.		in.	ga'ge.	corr.		ga'ge.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
13	10 min.	1	⅞	74.1	72.70	141	69.4	72.7	0.9	0.85	0.05	0.05	0.80	1.60	0.04	0.02	6½	1.47	0.03	0.01	9.22	1.30	0.56	0.06	0.31	0.25	81	
14	" "	1	1⅛	44.6	43.20	247	49.9	53.2	2.9	2.70	0.15	0.14	2.56	2.59	0.12	0.05	" "	2.57	0.10	0.04	16.11	4.03	1.76	0.15	0.99	0.84	85	
15	" "	1	1½	56.	54 60	344	68.2	71.5	5.2	4.87	0.35	0.33	4.54	3.90	0.23	0.10	" "	3.59	0.20	0.09	22.46	7.83	3.42	0.34	1.55	1.21	78	
16	" "	1	1⅝	29.5	28.30	497	72.0	75.3	11.2	10.41	1.10	1.02	9.39	5.64	0.50	0.22	" "	5.20	0.42	0.18	32.50	16.38	7.10	1.06	3.53	2.47	70	
17	" "	1	1⅝	20.2	19.00	407	41.7	45.0	7.5	6.99	0.7	0.64	6.35	4.61	0.33	0.14	" "	4.25	0.28	0.12	26.57	11.01	4.78	0.66	2.35	1.69	72	
18	" "	2	1½	66.2	64.80	375	67.5	70.8	2.0	1.87	0.5	0.47	1.40	4.24	0.28	0.12	" "	3.91	0.24	0.10	12.25	2.33	1.01	0.49	0.98	0.49	50	
19	" "	2	1⅝	29.8	28.60	500	37.3	40.6	3.4	3.16	0.94	0.87	2.29	5.68	0.50	0.2.	" "	5.23	0.42	0.19	16.32	4.16	1.81	0.90	1.57	0.67	43	
20	" "	2	1⅝	45.6	44.40	621	58.3	61.6	5.4	5.02	1.39	1.30	3.72	7.04	0.77	0.33	" "	6.49	0.66	0.29	20.27	6.40	2.78	1.34	2.57	1.23	48	
21	" "	2	2	34.4	33.20	717	53.2	56.5	7.2	6.70	1.85	1.72	4.98	8.14	1.03	0.44	" "	7.50	0.87	0.38	23.39	8.50	3.69	1.79	3.45	1.67	48	
22	" "	2	2½	17.4	16.20	832	49.5	52.8	10.1	9.42	2.94	2.73	6.69	9.45	1.39	0.60	" "	8.71	1.18	0.51	27.15	11.50	5.00	2.82	5.02	2.20	44	

Zero corr. = 1.4 and 1.2.

EXPERIMENTS ON HOLYOKE COMPRESSION (No. 1) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 42°. Temperature of Atmosphere, 45°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
									Measured Loss between 6″ Piezometer and															Act'l Frict'n Losses.				
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		2½″ Piezometer. (Lbs.)		Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		From 6″ Piezr to Gauge Connection at Top of Hydrant Bbl. (9) − [(15)−(12)].	From 6″ to 2½″ Piezrs Equals the Total Loss. (8)−(17)−(12).	From Top of Hyd. Bbl. to 2½″ Piezr. (19)−(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Nov. 12 & 13, 1897.		in.	g'e.	cor.		g'e.	cor	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
73	10 min.	1	¼	65.5	54.1	132	64.4	67.7	0.75	0.70	.05	0.05	0.65	1.50	0.03	0.01	6¼	1.38	0.03	0.01	8.63	1.16	0.50	0.05	0.21	0.16	76	
74	" "	1	1¼	46.0	44.6	250	51.9	55.2	2.80	2.60	.25	0.22	2.38	2.83	0.12	0.05	" "	2.61	0.11	0.05	16.31	4.15	1.81	0.22	0.84	0.62	74	
75	" "	1	1½	51.8	50.4	332	65.1	68.4	4.62	4.30	.40	0.37	3.93	3.76	0.22	0.10	" "	3.46	0 19	0.08	21.67	7.30	3.18	0.35	1.22	0.83	67	
76	" "	1	1⅝	23.8	22.6	444	54.4	57.7	8.72	8.10	.80	0.75	7.35	5.02	0.40	0.17	" "	4.62	0.33	0.14	28.98	13.09	5.68	0.78	2.59	1.81	70	
77	" "	1	2	17.0	15.8	495	59.9	63.2	11.49	10.67	1.00	0.93	9.74	5.62	0.49	0.21	" "	5.18	0.42	0.18	32.37	16.25	7.05	0.96	3.83	2.87	75	
78	" "	2	1½	62.9	61.5	365	64.6	67.9	1.90	1.77	.50	0.47	1.30	4.13	0.27	0.12	" "	3.82	0.23	0.10	11.93	2.20	0.97	0.40	0.92	0.43	47	
79	" "	2	1⅝	30.4	29.2	504	40.4	43.7	3.57	3.31	1.00	0.93	2.38	5.73	0.51	0.22	" "	5.28	0.43	0.19	16.44	4.22	1.82	0.96	1.70	0.74	43	
80	" "	2	1¾	42.8	41.6	602	58.0	61.3	5.04	4.71	1.40	1.30	3.41	6.83	0.73	0 32	" "	6.29	0.62	0.27	19.66	6.00	2.61	1.35	2.42	1.07	40	
81	" "	2	2	31.1	29.9	680	53.6	56.9	6.57	6.09	1.80	1.67	4.42	7.72	0.93	0.40	" "	7.11	0.79	0.34	22.19	7.65	3.33	1.73	3.15	1.43	45	
82	" "	2	2½	17.6	16.4	837	45.6	48.9	10.32	9.60	2.80	2.60	7.00	9.50	1.40	0.61	" "	8.75	1.19	0.52	27.31	11.64	5.05	2.69	5.15	2.47	48	

EXPERIMENTS ON HOLYOKE COMPRESSION (NO. 2) HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 25°.

1	2	3	4	5		6	7		8		9		10	11	12	13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6" Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6" Piezometer and 2½" Piezometer. (Lbs.)		Measured Loss between 6" Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½" Piezom'r. (8) – (9)	Velocity at 6" Piezometer.	Head Correspond'g to Column 11.	Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½" Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6" Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9) – (15) – (12).	From 6" to 2½" Piez'rs Equals the Total Loss. (8) – (17) – (12).	From Top of Hyd. Bbl. to 2½" Piez'r. (19) – (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Jan. 3. m.	s.	in.	g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft. lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.	50	
805	5	35½	1	¾	66.25	65.80	134	64.4	67.7	0.8	0.74	0.15	0.13	0.61	1.52	0.03	0.0	5¼	1.73	0.0.	0.0.	8.77	1.18	0.51	0.12	0.24	0.12	58	
806	5	44½	1	1¼	46.13	45.68	253	52.7	56.0	3.07	2.84	0.5	0.46	2.3	2.87	0.13	0.0.	"	3.27	0.1.	0.0.	16.50	4.24	1.84	0.45	1.06	0.61	57	
807	6	30½	1	1¼	52.02	51.57	333	66.0	69.3	5.2	4.81	0.8	0.74	4.0	3.79	0.22	0.10	"	4.31	0.25	0.13	21.86	7.43	3.22	0.71	1.69	0.98	57	
808	6	32½	1	1½	24.34	23.92	457	59.0	62.3	9.77	9.04	1.50	1.39	7.67	5.17	0.42	0.18	"	5.88	0.54	0.19	29.86	13.87	6.02	1.38	3.20	1.82	60	
809	5	25	1	2	18.25	17.83	524	61.3	64.6	13.38	12.38	2.1	1.94	10.44	5.96	0.55	0.24	"	6.78	0.71	0.31	34.24	18.23	7.92	1.87	4.70	2.83		
810	5	33½	1	¾	66.3	65.85	134	65.1	68.4	0.83	0.77	0.15	0.13	0.6.	1.52	0.03	0.01	"	1.73	0.05	0.02	8.77	1.18	0.51	0.12	0.27	0.15	56	
811	5	44½	1	1¼	46.28	45.83	253	52.2	55.5	3.04	2.81	0.45	0.41	2.4	2.87	0.13	0.06	"	3.27	0.17	0.07	16.50	4.24	1.84	0.40	1.13	0.63	61	
812	6	31½	1	1½	54.03	53.58	341	66.7	70.0	5.2	4.81	0.8	0.74	4.0	3.86	0.23	0.10	"	4.39	0.30	0.13	22.27	7.70	3.34	0.71	1.57	0.86	55	
813	6	23½	1	1½	24.47	24.05	458	58.4	61.7	9.75	9.02	1.5	1.3	7.65	5.19	0.42	0.1.	"	5.91	0.54	0.19	29.92	13.94	6.06	1.38	3.14	1.76	56	
814	5	33½	1	2	17.73	17.31	517	61.3	64.6	12.93	11.95	2.0	1.8	10.1	5.87	0.53	0.23	"	6.68	0.69	0.30	34.80	17.75	7.70	1.78	4.48	2.70	60	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON HOLYOKE FOUR-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 34°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16		17	18	19	20	21
Test Number.	Date and Duration of Test.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2″ Piezom'r. (8) − (9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fric't'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9) − (15) − (12).	Act'l Fric't'n Losses. From 6″ to 2½″ Piez'r. Equals the Total Loss. (8) − (11) − (12).	Act'l Fric't'n Losses. From Top of Hyd. Bbl. to 2½″ Piez'r. (19) − (18).	Loss at Nozzle in Per Cent. of Total Loss. (2½) × 100 ÷ (19).	Remarks.
	Jan. 5. m.	s.	in.	g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.	
849	5	18½	4	2	20.24	19.82	552	23	25.3	1.5	1.39	0.3	0.28	1.11	6.25	0.61	0.26	7½	4.00	0.25	0.11	9.02	1.26	0.55	0.43	1.10	0.57	61	
850	5	57	4	2	36.18	35.76	740	42	45.3	2.65	2.45	0.53	0.49	1.96	8.40	1.10	0.48	"	5.38	0.45	0.19	12.10	2.27	0.98	0.78	1.95	1.17	60	
851	5	43½	4	2½	19.23	18.81	897	31	34.3	3.9	3.69	0.8	0.74	2.86	10.17	1.60	0.69	"	6.51	0.66	0.29	14.62	3.33	1.44	1.14	2.85	1.71	40	
852	5	29½	4	2½	27.61	27.19	1,080	43.7	47.0	5.6	5.18	1.04	0.96	4.22	12.26	2.33	1.01	"	7.85	0.96	0.42	17.63	4.83	2.10	1.55	4.09	2.54	62	
853	5	29	3	1¾	19.33	18.91	406	20	23.3	1.15	1.06	0.2	0.19	0.87	4.60	0.33	0.14	"	2.94	0.13	0.06	8.83	1.20	0.52	0.27	0.68	0.41	60	
854	4	59	3	1¾	41.18	40.76	596	45	48.3	2.4	2.22	0.31	0.29	1.93	6.76	0.71	0.31	"	4.33	0.29	0.13	13.00	2.62	1.13	0.47	1.40	0.93	66	
855	5	58½	3	2	36.2	35 78	742	45	48.3	3.74	3.46	0.55	0.51	2.95	8.42	1.10	0.48	"	5.39	0.45	0.19	16.12	4.03	1.75	0.80	2.19	1.39	63	
856	5	44½	3	2½	19.32	18.90	900	36	39.3	5.6	5.18	0.75	0.69	4.49	10.21	1.62	0.70	"	6.53	0.66	0.29	19.61	5.97	2.59	1.10	3.29	2.19	67	

Zero corr. = 0.42.

EXPERIMENTS ON HOLYOKE FOUR-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 46°. Temperature of Atmosphere, 64°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezom'r and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezom'r and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hyd't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fric't'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)—(15)—(12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(15)—(12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Jan. 5, m. s.		in.	ga'ge.	corr.		g'e.	cor.	ga'ge.	cor.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
857	7 30½	2	1¼	42.26	41.81	302	41.	44.3	1.15	1.06	0.1	0.10	0.96	3.42	0.18	0.08	7/8	2.19	0.07	0.03	9.88	1.50	0.65	0.15	0.49	0.24	69	
858	5 46	2	1⅛	30.35	29.93	510	37.5	40.8	3.37	3.12	0.2	0.19	2.93	5.79	0 51	0.22	"	3.71	0.21	0.09	16.65	4.32	1.87	0.32	1.47	1.15		
859	5 55½	2	1¼	45.3	44.88	626	57.3	60.6	5.00	4.62	0.33	0.30	4.32	7.10	0.79	0.34	"	4.54	0.32	0.14	30 43	6.50	2.82	0.50	2.14	1.64		
860	5 08	2	2	34.06	33.64	790	52.	55.3	6.7	6.20	0.51	0.47	5.73	8.18	1.03	0.45	"	5.24	0.43	0.19	23.50	8.59	3.72	0.73	2.93	2.20		
861	5 39½	1	1	65.53	65.08	134	63.	66.3	0.9	0.83	0.03	0.03	0.80	1.52	0.03	0.01	"	0.97	0.01	0.00	8.77	1.18	0.51	0.04	0.33	0.29	88	
862	5 48⅝	1	1⅛	46.14	45.69	253	51. 3	54.6	3.2	2.96	0.07	0.07	2.89	2.87	0.13	0.06	"	1.84	0.05	0.02	16.50	4.26	1.85	0.11	1.17	1.06	91	
863	6 39	1	1¼	53.05	52.60	338	65.5	68.8	5.4	5.00	0.12	0.11	4.89	3.83	0.23	0.10	"	2.45	0.09	0.04	22.05	7.58	3.29	0.17	1.81	1.64	91	
864	6 24½	1	1⅜	24.24	23 82	455	53.7	57.0	10.2	9.44	0.2	0.19	9.25	5.16	0.42	0.18	"	3.30	0.17	0.07	29.73	13.74	5.97	0.30	3.65	3.35	92	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON HOLYOKE SIX-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 46.° Temperature of Atmosphere, 64°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 8" Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6" Piezometer and 2½" Piezometer. (Lbs.)		Measured Loss between 6" Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½" Piezom'r. (8) – (9).	Velocity at 6" Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½" Piezometer.	Head Corresponding to Column 16.		Act'l Fricť n Losses. From 8" Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9) – (15) – (12).	From 6" to 2½" Piez'rs Equals the Total Loss. (8) – (17) – (12).	From Top of Hyd. Bbl. to 2½" Piez'r. (19) – (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	1898. June 2. m. s.		in.	g'ge.	cor.		g'ge.	cor	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	feet	lbs.	lbs.	lbs.	lbs.		
875	5 05	1	⅝	66.64	65.04	135	....	68	1.45	1.34	0.40	0.37	0.97	.....	.....	.....	9¼	1.23	0.02	0.01	16.30	4.15	1.87	0.07	0.92	0.85	92	
876	5 02	1	1¼	45.01	44.41	250	....	56	2.99	2.77	0.06	0.06	2.71	1.60	0.04	0.02	"	1.63	0.04	0.02	21.68	7.31	3.18	0.06	1.35	1.29	96	
877	5 09	1	1⅞	50.91	50.31	332	....	68	4.84	4.50	0.05	0.05	4.45	2.12	0.07	0.03	"	2.28	0.08	0.04	30.26	14.24	6.18	0.07	3.06	2.99	98	
878	5 01	1	1¼	25.07	24.52	463	....	64	9.88	9.18	0.05	0.05	9.13	2.96	0.14	0.06	"	2.53	0.10	0.04	33.40	17.33	7.52	0.08	4.25	4.17	98	
879	5 00	1	2	17.48	16.93	511	....	64	12.59	11.70	0.05	0.05	11.65	3.27	0.17	0.07	"											
880	4 59	Alternate. (2 2 2 2 2 2)	1¼	42.22	41.62	301	...	47	1.05	0.99	0.05	0.05	0.94	1.93	0.06	0.03	"	1.48	0.03	0.01	9.83	1.50	0.66	0.07	0.36	0.29	80	
881	4 57		1½	30.05	29.53	507	..	41	2.91	2.70	0.12	0.11	2.59	3.23	0.16	0.07	"	2.48	0.10	0.04	16.54	4.27	1.86	0.14	0.91	0.77	85	
882	4 59		1¼	45.04	44.52	623	..	62	4.44	4.11	0.18	0.16	3.95	3.98	0.25	0.11	"	3.06	0.15	0.07	20.33	6.42	2.80	0.20	1.42	1.22	86	
883	5 01		2	33.78	33.26	708	....	58	5.91	5.49	0.22	0.20	5.29	4.52	0.32	0.14	"	3.47	0.19	0.08	23.09	8.29	3.60	0.26	2.03	1.77	87	
884	5 01		2¼	16.04	15.52	807	....	53	7.67	7.11	0.30	0.28	6.83	5.15	0.41	0.18	"	3.96	0.24	0.10	26.33	10.81	4.69	0.36	2.60	2.24	86	

Zero corr. = 875 – 877 = 0.60
" = 878 – 879 = 0.55
" 880 = 0.60
" = 881 – 884 = 0.52

* See retests Nos. 1,197-1,225.

EXPERIMENTS ON HOLYOKE SIX-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 46°. Temperature of Atmosphere, 64°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 8″ Piezometer at Hydr't Entr'ce. (Lbs.)		Measured Loss between 6″ Piezometer and 24″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 24″ Piez'r. (8) – (9).	Velocity at 8″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydrant Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 24″ Piezometer.	Head Corresp'd'g to Column 16.		Act'l Fric't'n Losses. From 8″ Pi'z'r to Gauge Connection at Top of Hyd. Bbl. (9)—[(15)–(12)].	From 8″ to 24″ Piez'r Equals the Total Loss. (8) [(17) – (12)].	From Top of Hyd. Bbl. to 24″ Piez'r. (19) – (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	June 2, 1898. m. s.		in.	ga'ge.	corr.		g'e.	cor.	g'e.	cor.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
885	1 59½	3	2½	20.25	19.60	916	....	41	4.45	4.12	0.35	0.32	3.80	5.86	0.53	0.23	9¼	4.50	0.31	0.13	19.90	6.15	2.67	0.42	1.68	1.26	75	*
886	1 59½	3	2	35.90	35.35	738	....	51	2.73	2.54	0.25	0.23	2.31	4.71	0.35	0.15	" "	3.62	0.20	0.09	16.04	4.00	1.74	0.29	0.95	0.66	69	*
887	1 59½	3	1½	40.54	39.99	590	....	48	1.85	1.72	0.19	0.18	1.54	3.77	0.22	0.10	" "	2.90	0.13	0.06	12.88	2.57	1.12	0.22	0.70	0.48	69	*
888	2 59½	3	1¼	18.98	18.43	400	....	46	0.90	0.84	0.10	0.10	0.74	2.55	0.10	0.05	" "	1.96	0.06	0.03	8.68	1.27	0.52	0.12	0.37	0.25	68	*

Zero corr. = 885–888 incl. 0.55
" = 889 = 0.60
" = 892–893 = 0.55

* See retests Nos. 1197–1228.

EXPERIMENTS ON HOLYOKE SIX-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 52°. Temperature of Atmosphere, 76°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 8" Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6" Piezometer and 2½" Piezometer. (Lbs.)		Measured Loss between 6" Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 8" Piezom'r. (8) – (9).	Velocity at 8" Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection. *	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½" Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 8" Piez'r to Gauge Connection at Top of Hydrant Bbl. (9) – [(15) – (12)].	From 8" to 2½" Piez'rs Equals the Total Loss. 8 – [(17) – (12)].	From Top of Hyd. Bbl. to 2½" Piez'r. (19) – (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19.)	Remarks.
	June 21, 1898.																											
	m. s.		in.	g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft. lbs.		in.	f. p. s.	ft. lbs.		f. p. s.	ft. lbs.		lbs.	lbs.	lbs.		
1197	3 18½	1	½	65.81	65.31	134	....	68.2	0.90	0.83	0.18	0.16	0.67	0.86	0.01	0.00	9½	0.56	0.00	0.00	8.77	1.18	0.52	0.16	0.31	0.15	....	
1198	3 09½	1	¾	47.66	47.16	258	....	58.6	3.07	2.88	0.06	0.06	2.82	1.65	0.04	0.02	"	1.08	0.02	0.01	16.83	4.42	1.92	0.05	0.98	0.93	95	
1199	3 05½	1	1	52.01	51.51	334	....	68.5	5.00	4.67	0.03	0.03	4.64	2.14	0.07	0.03	"	1.40	0.03	0.02	21.80	7.40	3.23	0.02	1.47	1.45	99	†
1200	3 07½	1	1¼	25.08	24.61	463	....	63.5	9.80	9.10	0.05	0.05	9.05	2.96	0.14	0.06	"	1.94	0.06	0.03	30.26	14.23	6.19	0.08	2.97	2.89	97	
1201	3 05	1	2½	17.12	16.65	507	....	64.9	12.24	11.37	0.07	0.07	11.30	3.24	0.17	0.07	"	2.13	0.07	0.03	33.14	17.06	7.41	0.11	4.03	3.92	97	
1202	3 13½	2	1½	41.97	41.47	302	....	45.5	1.00	0.94	0.03	0.03	0.91	1.93	0.06	0.03	"	1.27	0.03	0.02	9.87	1.50	0.66	0.04	0.31	0.27	87	
1203	3 05½	2	1¾	28.81	28.34	494	....	49.2	2.78	2.59	0.06	0.06	2.53	3.14	0.15	0.07	"	2.06	0.07	0.03	16.12	4.03	1.76	0.10	0.90	0.80	89	
1204	5 02½	2	1¾	45.83	45.36	629	....	62.2	4.41	4.09	0.10	0.10	3.99	4.01	0.25	0.11	"	2.63	0.11	0.05	20.53	6.55	2.85	0.16	1.35	1.19	88	‡
1205	3 02	2 (Outlets Nos. 5& 1.	2	34.53	34.06	725	....	57.9	5.98	5.54	0.14	0.13	5.41	4.63	0.33	0.14	"	3.04	0.14	0.06	28.66	8.70	3.78	0.21	1.90	1.69	89	
1206	3 02½	2	2½	17.05	15.59	843	....	48.2	8.08	7.51	0.20	0.18	7.33	5.38	0.45	0.20	"	3.53	0.19	0.08	37.52	11.80	5.12	0.30	2.59	2.29	88	

Zero corr. = 0.47 and 0.50.

* Bbl. tapped at a lower point than previous tests causes difference in diameter.

† Bbl. loss too small to be measured accurately with the apparatus at hand.

‡ Alternate outlets with two piezometers.

EXPERIMENTS ON HOLYOKE SIX-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 52°. Temperature of Atmosphere, 76°.

1	2	3	4	5		6	7	8		9		10	11	12		13	14	15		16	17		18	19	20	21	
								Measured Loss between 6″ Piezometer and															Act'l Fric't'n Losses.				
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 8″ Piezometer at Hydrant Entr'nce (Lbs.)	2½″ Piezometer. (Lbs.)		Top of Hydrant Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8) – (9).	Velocity at 8″ Piezometer.	Head Corresponding to Column 11.		Diam'r of Hydrant Bbl at Gauge Connection.	Velocity in Hydrant Bbl at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		From 8″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9) – [(15) – (12)].	From 8″ to 2½″ Piez'rs Equals the Total Loss. (8) – [(17) – (12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19) – (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	1898. June 21. m. s.																										
				in.	g'ge	corr.		g'e.	cor.	g'e.	cor.	g'e.	cor.	lbs.	f. p. s.	ft	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
1207	3 03½	3	1¼	19.61	19.14	417	....	23.0	0.90	0.84	0.06	0.06	0.78	2.67	0.11	0.05	9½*	1.75	0.05	0.02	9.08	1.27	0.56	0.09	0.33	0.24	73	
1208	2 05½	3	1½	41.57	41.10	600	....	49.7	1.85	1.71	0.10	0.10	1.61	3.83	0.23	0.10	"	2.51	0.10	0.04	13.05	2.54	1.15	0.16	0.66	0.50	76	
1209	3 05½	3	2	38.10	37.63	761	....	51.6	3.09	2.89	0.19	0.17	2.72	4.87	0.37	0.16	"	3.20	0.16	1.07	16.55	4.27	1.86	0.20	1.19	0.93	78	
1210	3 04½	3	2½	20.68	20.21	930	....	41.7	4.48	4.16	0.27	0.24	3.92	5.95	0.55	0.24	"	3.90	0.24	0.10	20.23	6.36	2.77	0.38	1.63	1.25	88	
1211	3 07½	4	2	20.91	20.44	561	....	25.0	1.15	1.06	0.09	0.09	0.97	3.59	0.20	0.09	"	2.36	0.09	0.04	9.18	1.30	0.57	0.14	0.58	0.44	76	
1212	3 03½	4	2	36.34	35.87	745	....	45.7	2.00	1.88	0.16	0.14	1.74	4.75	0.35	0.15	"	3.12	0.15	0.07	12.17	2.30	1.00	0.22	1.03	0.81	79	
1213	3 03½	4	2½	18.96	18.49	889	....	32.4	2.81	2.60	0.20	0.18	2.42	5.68	0.50	0.22	"	3.73	0.24	0.10	14.50	3.28	1.43	0.30	1.39	1.09	78	
1214	3 01½	4	2½	26.90	25.53	1.023	....	45.2	3.94	3.66	0.30	0.27	3.39	6.53	0.66	0.29	"	4.29	0.29	0.13	16.68	4.34	1.88	0.43	2.07	1.64	79	
1215	3 14	2	1½	42.77	42.27	307	....	46.0	1.00	0.94	0.03	0.03	0.91	1.97	0.06	0.03	"	1.29	0.03	0.01	10.02	1.55	0.68	0.05	0.29	0.24	83	
1216	3 00½	2	1½	28.46	27.99	494	....	38.7	2.62	2.44	0.08	0.08	2.36	3.15	0.15	0.07	"	2.07	0.07	0.03	16.11	4.03	1.75	0.12	0.76	0.64	84	
1217	3 01½	2	1½	46.06	45.59	630	....	64.6	4.37	4.05	0.10	0.10	3.95	4.02	0.25	0.11	"	2.64	0.11	0.05	20.57	6.57	2.86	0.16	1.30	1.14	88	†
1218	3 03½	2	2	34.08	33.61	720	....	57.5	5.88	5.46	0.16	0.15	5.31	4.60	0.33	0.14	"	3.02	0.14	0.06	23.50	8.58	3.73	0.23	1.87	1.64	88	
1219	3 01	2	2½	17.33	18.86	849	....	51.2	7.98	7.40	0.20	0.18	7.22	5.41	0.45	0.19	"	3.55	0.20	0.09	27.70	11.98	5.20	0.28	2.39	2.11	88	

Zero corr. = 0.47 and 0.50. * Barrel tapped at a lower point than previous tests causes difference in diameter. † Adjacent outlets with one piezometer.

EXPERIMENTS ON HOLYOKE SIX-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 52°. Temperature of Atmosphere, 76°.

1	2	3	4	5		6	7	8		9		10	11	12		13	14	15		16	17		18	19	20	21	22	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 8″ Piezometer at Hyd. Entrance. (Lbs.)	Measured Loss between 6″ Piezometer and				Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)–(9).	Velocity at 8″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Correspond'g to Column 16.		Actual Friction Losses.			Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Totals of Column 6.	Remarks.
								2½″ Piezometer. (Lbs.)		Top of Hydr't Bbl. (Lbs.)													From 8″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)–(15)–(12).	From 8″ to 2½″ Piez'rs Equals the Total Loss. (8)–(17)–(12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)–(18).			
	June 21, 1898. m. s.		in.	g'ge.	corr.		g'e.	g'e.	cor.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.			
1220	2 30½	5	2	24.30	23.83	607	....	30.3	1.26	1.16	0.16	0.15	1.01	4.85	0.37	0.16	9¼	3.18	0.16	0.07	9.92	1.52	0.67	0.24	0.65	0.41	63	759	
			1	26.82	26.32	152																						
1221	2 24½	5	2	34.70	34.23	727	....	44.0	1.86	1.73	0.20	0.1	1.54	5.82	0.52	0.22	"	3.82	0.23	0.10	11.90	2.19	0.95	0.30	0.98	0.68	69	910	
			1	38.83	38.35	183																						
1222	2 31½	5	2	19.58	19.11	903	....	34.0	2.45	2.27	0.29	0.27	2.00	7.02	0.77	0.33	"	4.61	0.33	0.14	14.35	1.30	1.40	0.43	1.20	0.74	62	1100	
			1¼	27.80	27.30	197																						
1223	2 33½	5	2¼	23.33	22.86	989	....	41.3	3.41	3.17	0.38	0.35	2.82	7.94	0.98	0.42	"	5.21	0.42	0.18	16.21	4.10	1.78	0.59	1.81	1.22	67	1243	
			1	23.72	23.22	254																						
1224	2 31½	5	2	22.78	22.31	978	....	40.0	3.70	3.45	0.37	0.34	3.11	8.07	1.01	0.43	"	5.30	0.44	0.19	16.50	4.23	1.83	0.58	2.05	1.47	72	1264	
			1¼	28.45	24.95	286																						
			Nos. 1, 3, 4, 5 & 6—																									
1225	2 31	6	2½	19.38	18.91	900	....	34.0	2.48	2.30	0.38	0.35	2.95	8.24	1.05	0.45	"	5.41	0.46	0.20	14.06	3.08	1.34	0.60	1.41	0.81	57	1202	
			1¼			196																						
			1½	27.40	23.90	196																						
1226	2 31½	6	2½	21.03	20.56	938	....	36.8	2.71	2.51	0.44	0.40	2.11	8.59	1.15	0.50	"	5.63	0.49	0.21	14.64	3.33	1.46	0.69	1.55	0.86	55	1346	
			1¼			204																						
			1½	23.87	23.37	204																						
1227	2 31½	6	2½	20.03	19.56	915	....	35.0	2.96	2.74	0.47	0.43	2.31	8.85	1.21	0.52	"	5.81	0.52	0.22	15.08	3.54	1.53	0.73	1.73	1.00	58	1387	
			1¼			236																						
			1½	25.82	25.32	236																						
1228	2 30½	6	2½	11.92	11.45	693	....	20.5	1.80	1.67	0.29	0.2	1.40	6.81	0.72	0.32	"	4.49	0.31	0.13	11.60	2.08	0.91	0.46	1.08	0.62	57	1065	
			1¼			183																						
			1½	15.75	15.25	183																						

Zero corr. = 0.47 and 0.50. * For total quantities see column 22. † Bbl. tapped at a lower point than previous tests causes difference in diameter.

EXPERIMENTS ON LUDLOW 5-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 38°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fricť n Losses. From 6″ Piez'r to Gauge Connection at Top of Hydrant Bbl. (9)—[(15)—(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Dec. 14 and 15.																											
	m. s		in.	g'e.	cor.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
552	11 7	1	¼	66.8	66.4	135	65.0	68.3	0.80	0.74	0.10	0.10	0.64	1.53	0.03	0.01	7½	0.95	0.01	0.00	8.83	1.20	0.52	0.11	0.23	0.12	52	
553	11 47	1	1½	46.9	46.5	256	52.4	55.7	2.98	2.76	0.20	0.19	2.57	2.90	0.13	0.06	"	1.79	0.05	0.02	16.72	4.35	1.89	0.23	0.93	0.70	75	
554	11 28½	1	1½	53.7	53.3	340	66.0	69.3	5.12	4.78	0.30	0.28	4.50	3.86	0.23	0.10	"	2.39	0.09	0.04	22.30	7.66	3.33	0.34	1.55	1.21	78	
555	11 41½	1	1½	24.7	24.3	490	54.0	57.3	9.73	9.03	0.50	0.47	8.56	5.22	0.43	0.19	"	3.23	0.16	0.07	30.04	14.04	6.08	0.59	3.14	2.55	81	
556	10	1	2	19.0	18.6	535	62.8	66.1	13.71	12.73	0.75	0.69	12.04	6.08	0.58	0.25	"	3.76	0.22	0.10	35.02	19.07	8.28	0.84	4.70	3.86	82	
557	10 min.	2	1½	64.7	64.3	374	66.0	69.3	1.78	1.64	0.30	0.28	1.36	4.23	0.28	0.12	"	2.62	0.11	0.05	12.22	2.32	1.02	0.35	0.74	0.39	53	
558	"	2	1½	30.4	30.0	512	37.0	40.3	3.55	3.31	0.64	0.60	2.71	5.80	0.53	0.23	"	3.59	0.20	0.09	16.72	4.35	1.90	0.74	1.64	0.90	55	
559	"	2	1½	45.3	44.9	626	57.1	60.4	5.20	4.86	0.93	0.85	4.01	7.10	0.79	0.34	"	4.40	0.30	0.13	20.43	6.49	2.82	1.06	2.38	1.32	55	
570	"	2	2½	34.4	34.0	723	52.6	55.9	6.98	6.50	1.24	1.15	5.35	8.22	1.05	0.45	"	5.09	0.40	0.17	23.58	8.65	3.76	1.43	3.19	1.76	55	
571	"	2	2½	17.3	16.9	850	46.3	49.6	10.10	9.41	1.70	1.57	7.84	9.64	1.45	0.63	"	5.97	0.55	0.24	27.73	12.00	5.21	1.96	4.83	2.87	59	

Zero corr. = 0.4.

EXPERIMENTS ON LUDLOW 5-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 38°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hydrant Entr'ce. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydrant Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8)−(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydrant Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Col. 14		Velocity at 2½″ Piezometer.	Head Corresp'd'g to Column 16.		Act'l Fric't'n Losses. From 6″ Piez'r to Gauge Connection at Top of Hydrant Bbl. (9)−[(15)−(12)].	From 6″ to 2½″ Piez'r's Equals the Total Loss. (8)−[(17)−(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)−(18).	Loss at Nozzle in Per Cent. of Total Loss. (20)×100÷(19).	Remarks.
			in.	ga'ge.	corr.		g'e.	cor.	ga'ge.	cor.	ga'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
557	Dec. 14. 10 min.	1	¾	65.96	66.51	135	65.2	68.5	0.8	0.74	0.05	0.05	0.69	1.53	0.03	0.01	7.5	0.95	0.01	0.00	8.83	1.20	0.52	0.06	0.23	0.17	74	
558	" "	1	1¼	45.94	45.49	253	51.3	54.7	2.95	2.74	0.2	0.19	2.55	2.87	0.13	0.06	"	1.78	0.05	0.02	16.50	4.24	1.83	0.23	0.97	0.74	76	
559	" "	1	1½	53.07	52.62	338	65.0	68.3	5.00	4.65	0.3	0.28	4.37	3.83	0.23	0.10	"	2.37	0.09	0.04	22.06	7.57	3.29	0.34	1.46	1.12	77	
560	" "	1	1¾	24.44	24.02	458	53.9	57.2	9.69	9.00	0.5	0.47	8.53	5.19	0.42	0.18	"	3.21	0.16	0.07	29.92	13.92	6.03	0.58	3.15	2.57	82	
561	" "	1	2	18.9	18.48	534	63.8	67.1	13.66	12.67	0.754	0.70	11.97	6.06	0.57	0.25	"	3.75	0.22	0.10	34.89	18.92	8.21	0.85	4.71	3.86	82	
562	" "	1	1½	64.72	64.27	374	66.4	69.7	1.75	1.62	0.3	0.28	1.34	4.23	0.28	0.12	"	2.62	0.11	0.05	12.22	2.31	1.00	0.35	0.74	0.39	53	
563	" "	1	1½	30.48	30.06	512	38.6	41.9	3.85	3.57	0.636	0.61	2.96	5.80	0.53	0.23	"	3.59	0.20	0.09	16.72	4.35	1.90	0.75	1.90	1.15	61	
564	" "	1	1¾	15.76	45.34	628	57.7	61.0	5.05	4.73	0.936	0.87	3.86	7.14	0.79	0.34	"	4.42	0.30	0.13	20.50	6.53	2.84	1.08	2.23	1.15	52	
565	" "	1	2¼	34.4	33.98	724	52.0	55.3	6.85	6.37	1.28	1.18	5.19	8.22	1.05	0.45	"	5.09	0.40	0.17	23.62	8.68	3.77	1.46	3.05	1.59	52	
566	" "	1	2¼	17.33	16.91	850	45.1	48.4	9.67	9.00	1.7	1.56	7.44	9.64	1.45	0.63	"	5.97	0.55	0.24	27.73	12.00	5.21	1.95	4.42	2.47	56	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON MATHEWS 5-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. Temperature of Atmosphere, 58°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. En. trance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8) − (9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 14.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Actual Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)−(15) − (12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8) − (17) − (12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19) − (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Dec. 11.																										
	m.	s.																									
			in.	g'ge.	corr.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
509	11	11	1	⅝	66.08	65.63	134	64.2	67.5	0.85	0.78	0.1	0.10	0.68	1.52	0.03	0.01	7/16	0.99	0.02	0.01	8.77	1.18	0.51	0.10	0.28	0.18	64	
510	11	53½	1	1⅛	46.13	45.68	253	51.8	55.1	3.23	3.00	0.4	0.38	2.62	2.86	0.13	0.06	"	1.86	0.05	0.02	16.50	4.24	1.84	0.42	1.42	0.80	66	
511	11	32½	1	1⅛	53.36	52.91	339	66.1	69.4	5.2	4.87	0.7	0.64	4.23	3.84	0.23	0.10	"	2.50	0.10	0.04	22.12	7.61	3.32	0.70	1.65	0.95	58	
512	10	06½	1	1⅛	24.35	23.93	457	54	57.6	9.86	9.14	1.28	1.20	7.94	5.17	0.42	0.18	"	3.36	0.18	0.08	29.86	13.87	6.02	1.30	3.30	2.00	61	
513	10	02	1	2	18.76	18.34	530	61	64.9	13.94	12.94	1.75	1.62	11.32	6.01	0.56	0.24	"	3.91	0.24	0.10	34.62	18.63	8.08	1.76	5.10	3.34	65	
514	11	27½	2	1¼	65.7	65.25	376	67.1	70.4	2.27	2.10	0.8	0.73	1.37	4.25	0.28	0.12	"	2.77	0.12	0.05	12.28	2.33	1.02	0.80	1.20	0.40	33	
515	10	49	2	1⅜	30.02	29.60	508	36.9	40.2	4.08	3.89	1.48	1.38	2.51	5.78	0.52	0.22	"	3.76	0.22	0.10	16.57	4.28	1.86	1.50	2.25	0.75	33	
516	10	07⅔	2	1½	45.1	44.68	623	57.5	60.8	6.2	5.78	2.3	2.13	3.65	7.07	0.78	0.34	"	4.60	0.33	0.14	20.33	6.42	2.80	2.33	3.32	0.99	30	
517	10	52	2	2	34.39	33.97	724	53.3	56.3	8.41	7.83	3.1	2.91	4.92	8.22	1.05	0.45	"	5.35	0.45	0.20	23.62	8.67	3.77	3.16	4.51	1.35	30	
518	10	05	2	2½	17.6	17.18	858	47.5	50.8	11.57	10.75	4.37	4.04	6.71	9.76	1.47	0.64	"	6.35	0.63	0.27	28.00	12.22	5.31	4.41	6.08	1.67	27	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON MATHEWS 5-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 39°. Temperature of Atmosphere, 46°.

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)—(9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Friction Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)–[(15)—(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)—(17)—(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)—(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
			in.	ga'ge.	corr.		g'e.	cor.	ga'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
613	December 17, 1897. 10 minutes.	1	¾	66.07	65.62	134	64.5	67.8	0.85	0.78	0.10	0.10	0.00	1.52	0.03	0.01	7/16	0.99	0.02	0.01	8.76	1.19	0.52	0.10	0.27	0.17	63	
614		1	1¼	46.62	46.17	255	52.0	55.3	3.09	2.90	0.40	0.37	0.03	2.89	0.13	0.06	"	1.88	0.05	0.02	16.64	4.32	1.88	0.41	1.08	0.67	62	
615		1	1½	53.10	52.65	338	64.9	68.2	5.16	4.81	0.70	0.65	0.05	3.84	0.23	0.10	"	2.50	0.10	0.04	22.06	7.57	3.39	0.71	1.62	0.91	56	
616		1	1¾	24.18	23.76	455	51.6	54.9	9.76	9.06	1.27	1.17	0.10	5.16	0.42	0.18	"	3.36	0.18	0.08	29.72	13.54	5.88	1.27	3.36	2.09	62	
617		1	2	19.00	18.58	596	61.0	64.3	14.09	13.08	1.80	1.67	0.13	6.08	0.58	0.25	"	3.96	0.24	0.10	35.02	19.07	8.28	1.82	5.05	3.23	64	
618		2	1¼	65.78	65.33	377	65.7	69.0	2.20	2.04	0.80	0.74	0.06	4.26	0.28	0.12	"	2.77	0.12	0.05	12.32	2.34	1.01	0.81	1.15	0.34	30	
619		2	1½	30.41	29.99	512	37.9	41.2	4.15	3.87	1.54	1.43	0.11	5.80	0.53	0.23	"	3.77	0.22	0.10	16.72	4.35	1.90	1.56	2.20	0.64	29	
620		2	1¾	44.16	43.74	617	56.1	59.4	5.98	5.54	2.20	2.04	0.16	7.00	0.77	0.33	"	4.56	0.32	0.14	20.16	6.30	2.74	2.23	3.13	0.90	29	
621		2	2	33.97	33.55	722	52.4	55.7	8.29	7.71	3.02	2.83	0.19	8.19	1.04	0.45	"	5.33	0.44	0.19	23.57	8.64	3.75	3.09	4.41	1.32	30	
622		2	2	17.58	17.16	858	46.5	49.8	12.01	11.16	4.37	4.04	0.33	9.76	1.47	0.64	"	6.35	0.63	0.27	28.09	12 23	5.31	4.41	6.49	2.08	32	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON MATHEWS FOUR-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. **Temperature of Atmosphere, 25°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8)−(9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hyd'r't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fricт'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9)−[(15)−(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)−[(17)−(12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19)−(18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Jan. 3, 1898. m. s.		in.	g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
815	5 35⅔	1	¾	66.12	65.67	134	64.8	68.1	1.1	1.02	0.05	0.05	0.97	1.51	0.03	0.01	8¼	0.75	0.01	0.00	8.77	1.19	0.52	0.06	0.51	0.45	88	
816	5 46⅔	1	1¼	45.9	45.45	252	52.2	55.5	3.9	3.63	0.07	0.07	3.56	2.85	0.12	0.05	"	1.42	0.03	0.01	16.43	4.22	1.83	0.11	1.85	1.74	94	
817	6 37⅔	1	1½	53.1	52.65	338	66.3	69.6	6.61	6.14	0.15	0.13	6.01	3.83	0.23	0.10	"	1.91	0.06	0.03	22.07	7.57	3.29	0.20	2.95	2.75	93	
818	6 22	1	1¾	24.4	23.98	458	56.2	59.5	12.57	11.66	0.30	0.27	11.39	5.20	0.42	0.18	"	2.53	0.10	0.04	29.92	13.92	6.04	0.41	5.80	5.39	93	
823	7 29½	2	1½	42.05	41.60	301	41.0	44.3	1.4	1.28	0.15	0.13	1.13	3.40	0.18	0.08	"	1.69	0.04	0.02	9.87	1.51	0.66	0.19	0.70	0.51	73	
820	5 50⅔	2	1½	30.32	29.90	510	37.6	40.9	4.02	3.75	0.40	0.36	3.39	5.79	0.52	0.22	"	2.88	0.13	0.06	16.64	4.32	1.88	0.52	2.09	1.57	75	
821	5 57	2	1½	45.43	45.01	627	58.0	61.3	6.08	5.66	0.60	0.54	5.12	7.12	0.79	0.34	"	3.55	0.20	0.09	20.47	6.47	2.82	0.79	3.18	2.39	75	
822	5 07⅔	2	2	34.40	33.98	724	53.5	56.8	8.13	7.57	0.73	0.67	6.90	8.22	1.05	0.45	"	4.10	0.26	0.11	23.61	8.68	3.77	1.01	4.25	3.24	76	

Zero corr. = 0.42 and 0.45.

EXPERIMENTS ON MATHEWS FOUR-WAY HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897-8.)

Temperature of Water, 38°. **Temperature of Atmosphere, 14°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8) − (9).	Velocity at 6″ Piezometer.	Head Corresponding to Column 11.		Diameter of Hydr't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresponding to Column 14.		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fric't'n Losses. From 6″ Piez'r to Gauge Connection at Top of Hydrant Bbl. (5) − [(15) − (12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8) − [(17) − (12)].	From Top of Hyd. Bbl. to 2½″ Piez'r. (19) − (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Jan. m.	d. h. m. s.	in.	g'ge.	corr.		g'ge.	cor.	g'ge.	corr.	g'ge.	cor.	lbs.	f. p. s.	ft.	lbs	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
824	6	31½	3	1¾	24.25	23.83	455	26.0	29.3	1.66	1.53	0.30	0.27	1.26	5.15	0.42	0.18	8½	2.57	0.10	0.04	9.92	1.53	0.67	0.41	1.04	0.63	61	
825	5	25	3	1¾	35.14	34.72	550	38.1	41.4	2.33	2.16	0.40	0.36	1.80	6.23	0.60	0.26	" "	3.10	0.15	0.07	12.00	2.22	0.97	0.55	1.45	0.90	62	
826	5	58	3	2	36.48	36.06	747	46.0	49.3	4.36	4.04	0.70	0.63	3.41	8.50	1.12	0.49	" "	4.24	0.28	0.12	16.24	4.11	1.79	1.00	2.74	1.74	63	
827	5	45½	3	2¼	19.33	18.91	900	36.0	39.3	6.30	5.87	1.10	1.01	4.86	10.21	1.62	0.70	" "	5.09	0.40	0.17	19.60	5.97	2.59	1.54	3.98	2.44	61	
828	5	17½	3	2¼	22.73	22.31	977	42.8	46.1	7.43	6.90	1.20	1.10	5.80	11.07	1.91	0.83	" "	5.52	0.47	0.20	21.25	7.04	3.07	1.73	4.66	2.93	63	*
829	6	13½	4	2	23.33	22.91	595	26.5	29.8	1.98	1.83	0.4	0.37	1.46	6.76	0.71	0.31	" "	3.37	0.18	0.08	9.75	1.47	0.64	0.60	1.50	0.90	60	
830	5	58	4	2	36.09	35.67	743	41.7	45.0	3.07	2.87	0.7	0.65	2.22	8.44	1.11	0.48	" "	4.21	0.27	0.12	12.15	2.29	1.00	1.01	2.35	1.34	57	
831	5	38½	4	2¼	26.76	26.34	1064	44.2	47.5	6.4	5.96	1.4	1.30	4.66	12.07	2.27	0.98	" "	6.01	0.56	0.24	17.37	4.70	2.04	2.04	4.90	2.86	58	*
832	5	47½	4	2¼	19.28	18.86	900	31.6	34.9	4.55	4.23	1.0	0.93	3.30	10.21	1.62	0.70	" "	5.09	0.40	0.17	14.67	3.37	1.47	1.46	3.46	2.00	58	

Zero corr. = 0.42. * All pressure available.

EXPERIMENTS ON PRATT & CADY 6-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. **Temperature of Atmosphere, 39°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entr'ce. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hyd. Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezometer. (8)–(9).	Velocity at 6″ Piezometer.	Head Corresp'd'g to Column 11.		Diameter of Hydrant Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Corresp'd'g to Column 14		Velocity at 2½″ Piezometer.	Head Corresponding to Column 16.		Act'l Fric't'n Losses. From 6″ Piez'r to Gauge Connec'n at Top of Hydrant Bbl. (9)–[(15)–(12)].	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8)–[(17)–(12)].	From Top of Hyd'l Bbl. to 2½″ Piez'r. (19)–(18).	Loss at Nozzle in Per Cent. of Total Loss. (20)÷(19)÷19.	Remarks.
	Dec. 13. m. s.	in.		ga'ge.	cor.		g'e.	cor.	ga'ge.	cor.	ga'ge.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
519	11 11	1	¼	66.1	65.65	134	64.5	67.8	0.8	0.74	0.05	0.05	0.69	1.52	0.03	0.01	6	1.52	0.03	0.01	8.77	1.18	0.51	0.05	0.24	0.19	79	
520	11 49½	1	1⅛	46.34	45.89	254	52.3	55.6	2.8	2.60	0.05	0.05	2.55	2.88	0.13	0.06	"	2.88	0.13	0.06	16.58	4.28	1.87	0.05	0.79	0.74	94	
521	11 30⅝	1	1½	53.65	53.20	340	65.8	69.1	4.8	4.47	0.15	0.14	4.33	3.85	0.23	0.10	"	3.85	0.23	0.10	22.20	7.66	3.33	0.14	1.24	1.10	89	
522	11 47½	1	1⅛	24.05	23.63	454	51.8	55.1	8.9	8.27	0.25	0.24	8.03	5.15	0.42	0.18	"	5.15	0.42	0.18	29.65	13.69	5.94	0.24	2.51	2.27	90	
523	11 28⅘	1	2	18.7	18.28	582	60.3	63.6	12.61	11.69	0.35	0.33	11.36	6.03	0.57	0.25	"	6.03	0.57	0.25	34.76	18.78	8.15	0.33	3.79	3.46	91	
	Dec. 14.																											
534	11 31⅔	2	1½	65.04	64.62	375	66.0	69.3	1.7	1.57	0.20	0.19	1.38	4.24	0.28	0.12	"	4.24	0.28	0.12	12.22	2.33	1.01	0.19	0.68	0.49	72	
535	10 33⅝	2	1½	30.54	30.12	512	36.8	40.1	3.19	3.00	0.30	0.28	2.72	5.80	0.53	0.23	"	5.80	0.53	0.23	16.72	4.35	1.90	0.28	1.33	1.05	79	
536	10 12⅝	2	1¼	44.09	43.67	617	55.2	58.5	4.61	4.29	0.50	0.47	3.82	7.00	0.77	0.33	"	7.00	0.77	0.33	20.16	6.30	2.74	0.47	1.88	1.41	75	
537	11 04⅝	2	2	33.2	32.78	711	50.3	53.6	6.2	5.77	0.68	0.63	5.14	8.07	1.01	0.44	"	8.07	1 01	0.44	23.20	8.37	3.63	0.63	2.58	1.95	76	
537	10 14	2	2½	17.19	16.77	847	43.6	46.9	9.0	8.37	0.94	0.87	7.50	9.62	1.43	0.62	"	9 62	1.43	0.62	27.64	11.92	5.17	0.87	3.82	2.95	77	

Zero corr. = 0.42 and 0.43.

EXPERIMENTS ON PRATT & CADY 6-INCH HYDRANT.

(Experiments of Chas. L. Newcomb, for City of Holyoke, 1897–8.)

Temperature of Water, 38°. **Temperature of Atmosphere, 40°.**

1	2	3	4	5		6	7		8		9		10	11	12		13	14	15		16	17		18	19	20	21	
Test Number.	Date and Duration of Tests.	No. of Outlets Open.	Diameter of Meter Nozzle.	Pressure at Meter Nozzle. (Lbs.)		Gallons per Minute by Nozzle.	Pressure on 6″ Piezometer at Hyd. Entrance. (Lbs.)		Measured Loss between 6″ Piezometer and 2½″ Piezometer. (Lbs.)		Measured Loss between 6″ Piezometer and Top of Hydr't Bbl. (Lbs.)		Loss between Hydrant Bbl. and 2½″ Piezom'r. (8) − (9).	Velocity at 6″ Piezometer.	Head Correspond'g to Column 11.		Diameter of Hyar't Bbl. at Gauge Connection.	Velocity in Hydrant Bbl. at Gauge Connection.	Head Correspond'g to Column 14.		Velocity at 2½″ Piezometer.	Head Correspond'g to Column 16.		Act'l Fricт'n Losses. From 6″ Piez'r to Gauge Connect'n at Top of Hyd. Bbl. (9) − (15) − (12).	From 6″ to 2½″ Piez'rs Equals the Total Loss. (8) − (17) − (12).	From Top of Hyd. Bbl. to 2½″ Piez'r. (19) − (18).	Loss at Nozzle in Per Cent. of Total Loss. (20) × 100 ÷ (19).	Remarks.
	Dec. 13 m. s.		in.	g'ge.	corr.		g'e.	cor.	g'ge.	corr.	g'e.	cor.	lbs.	f. p. s.	ft.	lbs.	in.	f. p. s.	ft.	lbs.	f. p. s.	ft.	lbs.	lbs.	lbs.	lbs.		
524	11 15	1	⅞	65.18	64.73	133	63.8	67.1	0.795	0.74	0.05	0.05	0.69	1.51	0.03	0 01	6	1.51	0.03	0.01	8.66	1.17	0.51	0.05	0.24	0.19	79	
525	11 52½	1	1⅛	46.4	45.95	254	51.5	54.8	2.86	2.65	0.10	0.10	2.55	2.88	0.13	0.06	"	2.88	0.13	0.06	16.58	4.28	1.87	0.10	0.84	0.74	88	
526	11 33	1	1⅛	53.48	53.03	340	55.5	68.8	4.76	4.43	0.20	0 19	4.24	3.86	0.23	0.10	"	3.86	0.23	0.10	22.20	7.66	3.33	0.19	1.20	1.01	84	
527	10 0⁵⁹⁄₆₀	1	1½	24.25	23.84	455	52.0	55.3	9.00	8.38	0.30	0.28	8.10	5.16	0.42	0.18	"	5.16	0.42	0.18	29.72	13.54	5.88	0.28	2.68	2.40	90	
528	11 1⁵⁸⁄₆₀	1	2	19.09	18.6.	538	62.5	65.8	12.95	12.02	0.40	0.37	11.65	6.11	0.58	0.27	"	6.11	0 58	0.25	35.15	19.20	8.39	0.37	3.88	3.51	90	
529	10 32⅝	2	1½	64 98	64.53	374	36.3	69.6	1.7	1.57	0 20	0.19	1.38	4.23	0 28	0.12	"	4.23	0 28	0.12	12.22	2.32	1.02	0.19	0.67	0.48	72	
530	10 42	2	1½	39.32	29.90	510	36.8	40.1	3.18	2.98	0.30	0.28	2.70	5.79	0.52	0.22	"	5.79	0.52	0.22	16.64	4.31	1.87	0.28	1.33	1.05	79	
531	10 05⅝	2	1½	45.49	45.07	627	56.6	59.9	4.71	4.38	0.50	0.47	3.91	7.12	0.79	0.34	"	7.12	0.79	0.34	20.48	6.51	2.84	0.47	1.88	1.41	75	
532	10 52⅝	2	2	34.55	34.13	726	52.6	55.9	6.49	6.02	0.70	0.64	5.38	8.23	1.05	0.45	"	8.23	1.05	0.45	23.68	8.72	3.79	0.64	2.68	2.04	76	
583	10 04½	2	2¼	17.70	17.28	860	46.0	49.3	9.3	8.64	0.93	0.85	7.79	9.77	1.48	0.64	"	9.77	1.48	0.64	28.07	12.28	5.33	0.85	3.95	3.10	78	

Zero corr. = 0.42 and 0.45.

TABULATED RESULTS OF OPEN-BUTT TESTS.

1	2	3	4	5	6	7	8	9	10	11	12	13
Test Number.	Date and Duration of Test.	Name of Hydrant.	Name of Meter.	Outlets Open.	Diameter of Hydrant Outlets. (Ins.)	Mercury Column Readings. (Lbs.)	Correction for Mercury Column. (Elevation.) (Lbs.)	Pressure at Back of Hydrant Outlet. (7 − 8.)	Total Cubic Feet Passing by Meter.	Gallons per Minute by Meter.	Gallons per Minute Corrected for Meter Error.	Discharge if Outlets had been Exactly 2½ Ins. (G. P. M.)
	Dec. 6.							lbs.				
404	6 34		Worthington.			13.28	2.83	10.45	400	455.91	475	459
405	5 27½		"			18.17	"	15.34	400	548.66	576	556
406	5 56¾		"	One, left hand.	L. H. 2.543 R. H. 2.518	23.40	"	20.57	500	628.92	663	640
407	5 23⅝		"			27.77	"	24.94	500	694.30	734	709
408	5 53		"			33.48	"	30.65	600	762.81	808	780
409	5 31⅞		"			38.04	"	35.21	600	812.03	859	830
410	5 06¾		"			44.15	"	41.32	600	878.23	933	902
411	5 04⅝		"		L. H. 2.543	13.34	"	10.51	600	884.6	940	918
412	5 00		"	Two.	R. H. 2.518	18.21	"	15.37	700	1,047.2	1,115	1,088
413	5 03		"		Aver. 2.5305	22.28	"	19.45	800	1,185	1,264	1,232
414	5 43⅛		"			13.20	2.73	10.47	400	522.5	547	547
415	5 56⅜		"			18.06	"	15.33	500	629.6	664	664
416	5 13⅜		"	One, right hand.	L. H. 2.518 R. H. 2.5	23.04	"	20.31	500	715.6	758	758
417	5 39⅝		"			27.84	"	25.11	600	792.9	840	840
418	5 14⅛		"			32.42	"	29.69	600	857.0	908	908
419	5 31⅜		"			38.76	"	36.03	700	938.9	998	998
420	5 19½		"			41.95	"	39.22	700	983.0	1,047	1,047
	Dec. 7.											
421	5 36¼		"			13.33	2.83	10.50	400	533.0	558	547
422	5 43⅜		"			18.57	"	15.74	500	653.5	690	676
423	5 53		"	One, left hand.	L. H. 2.525 R. H. 2.524	23.64	"	20.81	500	734.3	777	761
424	5 32⅜		"			28.10	"	25.27	600	809.1	857	841
425	5 07⅜		"			32.94	"	30.11	600	876.0	930	912
426	5 28⅝		"			39.26	"	36.43	700	956.0	1,017	997
427	5 18⅛		"		L. H. 2.525							
428	5 28⅜		"	Two.	R. H. 2.524	13.16	"	10.33	700	987.3	1,051	1,025
					Aver. 2.5245	18.95	"	16.12	900	1,229.2	1,311	1,289
429	5 47⅜		"			13.20	2.80	10.40	400	516.8	542	532
430	6 02⅜		"			18.13	"	15.33	500	619.5	653	642
431	5 12		"			23.30	"	20.50	500	719.2	760	747
432	5 40⅜		"	One, left hand.	L. H. 2.522 R. H. 2.522	28.13	"	25.33	600	791.1	838	824
433	5 13		"			33.18	"	30.38	600	860.3	913	898
434	5 38⅜		"			38.25	"	35.45	700	928.4	987	970
435	5 26¼		"			41.76	"	38.96	700	963.1	1,025	1,008
436	5 33⅛		"		L. H. 2.522	13.47	"	10.67	800	1,077.5	1,148	1,128
437	5 19⅜		"	Two.	R. H. 2.522	17.89	"	15.09	900	1,263.8	1,349	1,325
438	5 24⅜		"			12.68	2.45	10.23	300	415.0	430	427
439	5 55		"			17.43	"	14.98	400	505.7	530	526
440	5 07		"			22.61	"	20.16	400	584.0	614	610
441	5 48⅜		"	One, left hand.	L. H. 2.508 R. H. 2.515	27.36	"	24.91	500	644.1	680	675
442	5 15⅝		"			32.53	"	30.08	500	711.0	752	747
443	5 54		"			37.59	"	35.14	600	760.2	804	798
444	5 29⅜		"			43.07	"	40.62	600	817.0	865	860
	Dec. 8.				L. H. 2.508							
447	5 42		"	Two.	R. H. 2.515	12.66	"	10.21	700	907.9	964	956
448	5 11⅜		"		Aver. 2.5115	18.19	"	15.74	800	1,120.9	1,195	1,185
460	5 27		"			13.18	3.11	10.07	400	548.3	575	579
461	5 35⅜		"			18.16	"	15.05	500	669.5	707	712
462	5 49⅝		"	One, left hand.	L. H. 2.49 R. H. 2.51	23.31	"	20.20	600	769.8	815	821
463	5 17⅜		"			28.30	"	25.19	600	847.3	898	905
464	5 41⅜		"			33.37	"	30.16	700	919.7	978	986
465	5 28⅜		"			35.69	"	32.58	700	956.0	1,017	1,025
466	5 00⅜		"		L. H. 2.49.							
467	5 32⅜		"	Two.	R. H. 2.51	13.62	"	10.51	700	1,044.4	1,112	1,112
					Aver. 2.5	17.47	"	14.36	900	1,215.9	1,297	1,297
744	5 32½		Neptune.			13.21	2.8	10.41	400	539	530	521
745	5 44⅛		"			18.29	"	15.49	500	652	642	631
747	5 25⅜		"	One, left hand.	L. H. 2.522 R. H. 2.522	28.2	"	25.4	600	828	818	805
748	5 47⅛		"			32.99	"	30.19	700	905	895	880
749	5 22⅝		"			37.93	"	35.13	700	973	962	945
750	5 03⅜		"			42.93	"	40.13	700	1,035	1,022	1,006
751	5 35		"			45.8	"	43.00	800	1,072	1,058	1,040
752	5 34⅜		"			13.35	"	10.55	900	1,206	1,191	1,171
753	5 23		"	Two.		18.35	"	16.55	1,000	1,389	1,371	1,350
754	5 20		"			22.11	"	19.31	1,100	1,543	1,524	1,500
755	5 03½		"			13.33	3.11	10.22	400	592	582	586
756	5 13⅜		"			18.16	"	15.05	500	716	706	711
757	5 26⅜		"			23.16	"	20.05	600	825	815	821
758	5 42⅜		"	One, left hand.	L. H. 2.49 R. H. 2.51	28.00	"	24.89	700	918	908	915
759	5 12		"			33.04	"	29.93	700	1,007	995	1,002
760	5 30⅜		"			38.15	"	35.04	800	1,086	1,072	1,080
761	5 20⅜		"			40.37	"	37.26	800	1,120	1,107	1,115
762	5 20⅜		"			13.54	"	10.43	800	1,120	1,107	1,115
763	5 32⅜		"	Two.		18.23	"	15.12	1,000	1,350	1,333	1,342
764	5 08⅛		"			20.49	"	17.38	1,000	1,456	1,438	1,448

TABULATED RESULTS OF OPEN-BUTT TESTS.

1	2	3	4	5	6	7	8		9	10	11	12	13
Test Number.	Date and Duration of Test.	Name of Hydrant.	Name of Meter.	Outlets Open.	Diameter of Hydrant Outlets. (Ins.)	Mercury Column Readings. (Lbs.)	Corrections for Mercury Column. a U-tube (Corrected). (Lbs.)	Corrections for Mercury Column. b Elevation. (Lbs.)	Pressure at Back of Hydrant Outlet. (7 − 8.) (Lbs.)	Total Cubic Feet Passing by Meter.	Gallons per Minute by Meter.	Gallons per Minute Corrected for Meter Error.	Discharge if Outlets had been Exactly 2½ ins. (G. P. M.)
	Nov. 26. m. | s.								lbs.				
292	5 | 38¼	Coffin Gate.	Worthington.	One, left hand.	L. H. 2.525	11.58	1.14	2.84	7.60	320	424.64	441	432
293	5 | 19	"	"	"	R. H. 2.525	11.18	1.01	"	7.33	300	422.02	438	429
294	5 | 36½	"	"	"		16.77	1.62	"	12.31	400	533.82	550	548
295	5 | 55⅝	"	"	"		22.44	2.27	"	17.33	500	631.01	665	652
296	5 | 14	"	"	"		28.24	2.86	"	22.54	500	714.64	756	741
297	5 | 39	"	"	"		34.63	3.67	"	28.12	600	794.32	841	825
298	5 | 17⅝	"	"	"		39.5	4.1	"	32.56	600	848.35	900	882
299	5 | 44⅝	"	"	"		45.35	4.76	"	37.75	700	911.66	969	950
300	5 | 39¼	"	"	"		46.88	4.85	"	39.19	700	926.17	985	965
301	3 | 18½	"	"	"		15.13	4.00	"	8.29	600	844.7	895	877
302	6 | 29⅜	"	"	Two.	L. H. 2.525	22.21	6.14	"	13.23	900	1,036.7	1,104	1,082
303	5 | 40	"	"	"	R. H. 2.525	27.28	7.88	"	16.56	900	1,188.0	1,268	1,243
304	5 | 41⅛	"	"	"	Aver. 2.525	27.97	7.92	"	17.21	900	1,183.8	1,263	1,239
305	5 | 40⅝	"	"	"		28.11	7.97	"	17.30	900	1,185.2	1,265	1,240
	Nov. 29.												
309	5 | 50⅜	Chapman, No. 1.	"	One, left hand.	L. H. 2.516	10.64	0.48	2.19	7.97	300	383.56	395	390
310	5 | 48	"	"	"	R. H. 2.500	10.68	0.48	"	8.01	300	386.9	398	393
311	6 | 00	"	"	"		16.49	0.75	"	13.55	400	498.67	521	514
	Nov. 30.												
312	5 | 12⅜	"	"	"		21.28	1.1	"	17.99	400	574.24	603	595
313	5 | 44⅜	"	"	"		26.45	1.32	"	22.94	500	651.10	686	677
314	5 | 12¼	"	"	"		32.6	1.71	"	28.70	500	718.77	760	750
315	5 | 52⅛	"	"	"		36.8	2.00	"	32.61	600	764.6	810	799
316	5 | 30⅜	"	"	"		42.7	2.20	"	38.31	600	815.0	863	851
317	5 | 17⅝	"	"	"		47.36	2.60	"	42.57	600	848.6	900	888
318	5 | 20⅛	"	"	"	L. H. 2.516	11.67	1.28	"	8.20	500	700.8	741	730
319	5 | 45	"	"	"	R. H. 2.5	17.61	2.22	"	13.30	700	910.6	967	951
320	5 | 44	"	"	"	Aver. 2.508	23.11	3.00	"	17.92	800	1,043.7	1,113	1,107
321	5 | 13⅛	"	"	"		27.15	3.61	"	21.45	800	1,146.4	1,222	1,215
322	6 | 19⅛	"	"	"		10.95	0.64	2.89	7.42	400	472.92	493	488
323	5 | 07⅝	"	"	"		16.32	0.92	"	12.51	400	584.37	614	607
324	5 | 20⅜	"	"	"		21.03	1.40	"	16.74	500	700.81	740	732
325	5 | 55⅜	"	"	One, left hand.	L. H. 2.512	26.5	1.61	"	22.00	600	757.25	802	794
326	5 | 04⅛	"	"	"	R. H. 2.512	32.72	2.28	"	27.55	600	884.62	938	928
327	5 | 45⅝	"	"	"		37.3	2.30	"	32.11	700	898.62	954	945
328	5 | 26⅜	"	"	"		42.33	2.80	"	36.74	700	962.50	1,025	1,015
329	5 | 45⅝	"	"	"		38.1	2.48	"	32.73	700	909.03	966	957
330	5 | 36	"	"	"	L. H. 2.512	12.94	2.45	"	7.60	700	935.0	995	985
331	5 | 02⅜	"	"	Two.	R. H. 2.515	19.47	4.04	"	12.54	800	1,186.5	1,265	1,252
332	5 | 21⅛	"	"	"	Aver. 2.5135	21.46	4.54	"	14.03	900	1,255.3	1,340	1,326
	Dec. 1.												
342	6 | 46	Holyoke Gate, 6¼" bbl.	"	One, right hand.	R. H. 2.525	11.4	0.74	2.93	7.73	400	442.16	460	451
344	5 | 26½	"	"	"	L. H. 2.435	16.3	1.20	"	12.17	400	549.32	576	564
345	5 | 39⅜	"	"	"		22.35	1.74	"	17.68	500	660.39	697	683
346	5 | 17	"	"	"		27.41	2.10	"	22.38	500	707.9	749	734
347	5 | 33	"	"	"		33.38	2.80	"	27.65	600	808.65	856	840
348	5 | 05	"	"	"		39.27	3.35	"	32.99	600	882.88	938	921
349	5 | 42¼	"	"	"		44.64	3.51	"	38.20	600	918.10	976	957
350	5 | 12⅛	"	"	"	L. H. 2.437	13.65	2.94	"	7.78	600	862.5	916	931
351	5 | 34⅝	"	"	Two.	R. H. 2.525	20.42	4.85	"	12.64	800	1,073.7	1,145	1,162
352	5 | 00	"	"	"	Aver. 2.481	24.17	6.05	"	15.19	800	1,196.8	1,278	1,298
353	5 | 16⅜	Holyoke Compression, No. 1.	"	One, left hand.	L. H. 2.522	11.2	0.84	"	7.43	300	425.53	442	436
354	5 | 33⅛	"	"	"	R. H. 2.5215	16.45	1.28	"	12.24	400	538.7	565	556
355	5 | 52	"	"	"		22.48	1.87	"	17.68	500	637.5	673	663
356	5 | 11⅛	"	"	"		27.69	2.38	"	22.38	500	721.0	762	751
357	5 | 40	"	"	"		33.3	3.00	"	27.37	600	792.0	838	825
358	5 | 14⅜	"	"	"		39.06	3.35	"	32.78	600	855.49	907	894
359	5 | 42⅝	"	"	"		44.67	3.72	"	38.02	700	916.4	973	958
360	5 | 57	"	"	"		12.27	1.56	2.42	8.28	400	502.9	527	517
361	5 | 57⅜	"	"	"		17.65	2.32	"	12.91	500	628.2	662	650
362	5 | 26⅞	"	"	"		23.1	2.80	"	17.88	500	687.9	728	715
363	5 | 40⅞	"	"	One, left hand.	L. H. 2.524	29.42	3.68	"	23.32	600	790.6	837	823
364	5 | 15⅞	"	"	"	R. H. 2.523	35.13	4.25	"	28.46	600	853.2	905	889
365	5 | 02	"	"	"		40.38	4.57	"	33.39	600	891.6	946	929
366	5 | 31½	"	"	"		45.55	5.20	"	37.93	700	948.5	1,008	992
367	5 | 30½	Corey, 4".	"	Two.	L. H. 2.524	15.06	5.09	"	7.55	700	951.4	1,012	995
368	5 | 29⅝	"	"	"	R. H. 2.523	22.05	8.32	"	11.76	900	1,225.5	1,307	1,282
					Aver. 2.5235								

DCCCXII.*

THE MANUFACTURE OF CAR WHEELS.

BY G. R. HENDERSON, SCHENECTADY, N. Y.

(Member of the Society.)

LIKE other branches of mechanical industry, the manufacture of chilled iron car wheels has been a gradually progressing science. Fifteen years ago it was customary to purchase and supply wheels to railroads on a mileage guarantee, the only restrictions as to quality being the depth of chill and the size, shape, and weight of the wheel.

Certain mixtures were used by the founder which were known by experience to produce a given chill, and this and the color of the fracture, with perhaps the mileage records, were his guides in continuing his work. Chemical analyses of the pig iron and the finished output were considered unnecessary, if at all contemplated, and the irons, both coke and charcoal, were bought by fracture and chill. Some founders used a certain percentage of steel scrap, but this never became, as far as the writer knows, common practice.

Now all this is changed; the progressive wheelmaker buys his irons on specifications, and checks them by chemical analysis, and every heat is tested physically and chemically. Records are kept of the strength and resilience of the various irons on the market, and of the manner in which his wheels fail, and as many as possible of these are examined in order to avoid duplicating unsatisfactory products.

The actual shape of the wheel has undergone little change from the original Washburn section, except in the matter of size and weight, but we have every reason to believe that the older wheels cannot be compared with modern wheels in strength or endurance. A seeming inconsistency may be occasioned by a long life of old wheels, but it must be remembered that no such

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weight was then carried as is now the practice, where every effort is made in the interest of economical train movement.

Having been requested to speak on this subject, I will therefore endeavor to present some points about the manufacture of wheels which have come under my observation and which may at least lead to a discussion of this interesting question, to the mutual benefit of the members and the writer.

About the time when the drop test was first introduced the writer was assistant superintendent of a railroad shop which contained a foundry making 100 wheels per day. The introduction of this test, and the failure of a number of wheels in service about that time, caused him to make efforts to secure a stronger casting. Each day test bars were made along with the wheels, and these were subjected to transverse test, and a graphical record kept of the same, as shown by Fig. No. 214. The bars for this purpose were 2 inches square by 24 inches long, and were placed on supports $21\frac{1}{3}$ inches apart, and a centre load applied. As one-half the bars were chilled, it was considered that a 2-inch bar was a proper size to illustrate the chilling qualities, without becoming white through and through, and the span was fixed at $21\frac{1}{3}$ inches, so that when the proper deductions were made for the span and modulus of section the corresponding direct stress was exactly four times the centre load, thus facilitating the preparation of the form for recording the tests.

The bars of wheel iron so tested should stand a centre load of from 12,000 to 14,000 pounds before breaking, this corresponding to a maximum fibre strain of 48,000 to 56,000 pounds per square inch. The deflection at the moment of breaking is also of great importance, and there are wide differences between various irons in this respect. While the *strength* of the metal indicates a power to stand pressure, we need the *resilience* to withstand shocks. A ready means of comparing the relative resilience of different irons is derived from the curve shown in Fig. 214. As the ordinates represent loads and the abscissæ deflections, the area bounded by the axis of abscissæ, the ordinate passing through the final point of the curve (*i.e.* the ordinate corresponding to the final deflection), and the curve itself will represent the work done or absorbed by the test bar up to the breaking point; and for quick comparisons the small rectangles into which the sheet is divided may be counted within the

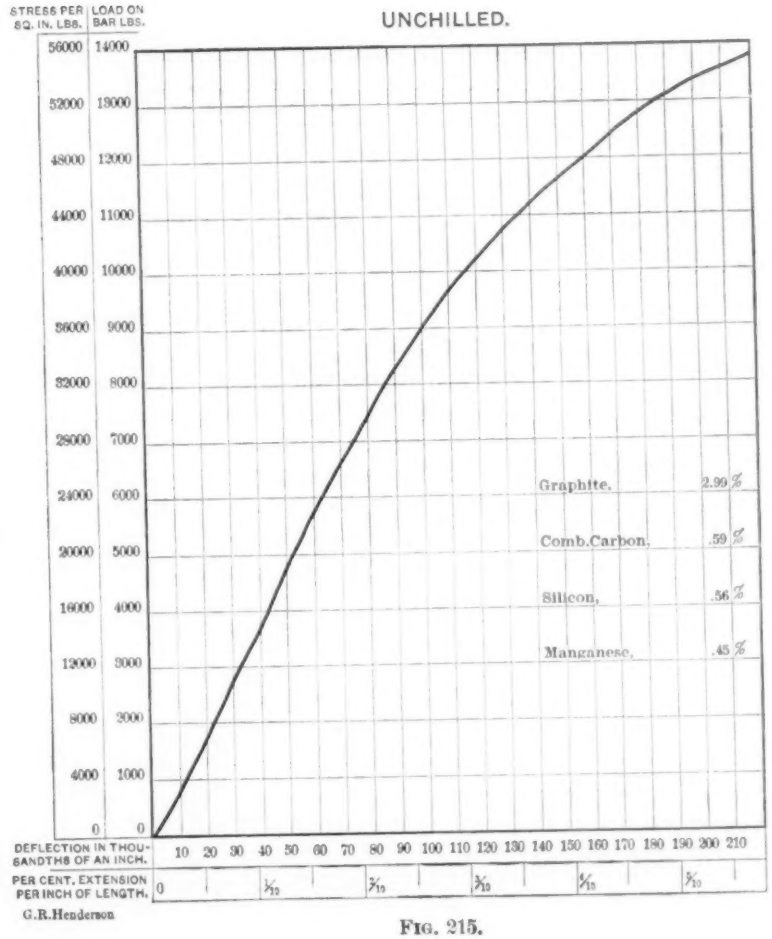


FIG. 215.

FIG. 214.

Norfolk and Western Railway Company.

MOTIVE POWER DEPARTMENT.

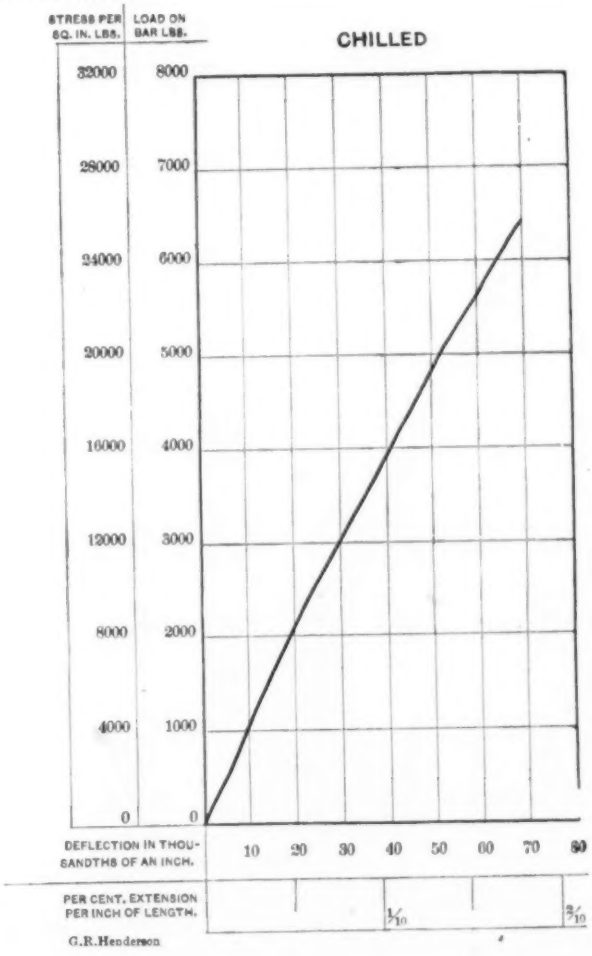
TRANSVERSE TEST OF CAST-IRON BARS.

ROANOKE, VA., DEC. 23, 1898.

DATA.

SIZE OF BAR, 2 × 2 INCHES. SPAN, 21½ INCHES.
SHRINKAGE IN 1 FOOT, ⅛ IN. FRACTURE, GRAY.
DEPTH OF CHILL, ⅜ INCH. DATE OF CAST, 11-22-98.

WHEEL IRON.....



boundaries above mentioned. Good wheel iron will give curves with an included area of from 150 to 200 of the small rectangles, and a total deflection of 0.2 inch at centre. Of course these comparisons can only be made with bars of the same size and under the same conditions.

This same treatment has been accorded to unmixed pig iron of the various grades suitable for wheel founding, the pig being melted separately in a small experimental cupola, and from these records the wheel founder makes his selections. The curves present to the eye a meaning easier to grasp than mere figures, and when bound together in a book comparisons can be quickly and accurately made. The amount of chill and shrinkage is also noted, together with the color of fracture and the chemical analysis of the iron.

As the composition of different shipments of iron from the same furnace will vary greatly, it naturally follows that the strength and chilling properties are not always the same.

It happened not long ago that a certain iron, which had previously given good results, became almost unfit for use, principally by reason of a great decrease in the amount of manganese. At one time this would probably not have been considered of such vital importance, but the fact has recently been well established that a certain amount of manganese is necessary in order that a wheel shall successfully stand the thermal test; it also seems to increase its strength under the drop test.

We have now seen that the wheel founder of the present day has the results of chemical analysis and physical tests of the unmixed iron and also of the mixture, as well as the results of the drop and thermal tests, to guide him in his work. The tests of to-day's heat are carefully studied by him in order that any defects may be corrected to-morrow. For this reason it is important that the analysis and deflection test should be placed in his hands early.

The drop and thermal tests cannot be made until the wheels are taken out of the annealing pits, a period of from five to eight days.

But this is not all. These tests are intended to show us quickly what kind of service we may expect from our wheels when under the cars, and the records of the life and failure of wheels in service are of the utmost importance. The actual mileage of a wheel under a freight car is a difficult thing to obtain; that is, it requires considerable labor, especially when

the number of wheels in use runs up into the hundred thousands. For this reason the life of wheels is often kept by months; and while this does not take into consideration the time spent in the repair yard or on the side track, it has the advantage of being very easily obtained.

As new wheels are made, the foreman of the foundry makes a daily report, giving the wheel numbers and the mixture used. The former are entered in a book which contains the serial wheel numbers, and the latter are entered in another book which later has the result of the drop and thermal tests recorded, as well as the depth of chill, shrinkage, etc. When the wheels are mounted on the axles a report is made weekly of the numbers and place at which this work was done, and this is entered in the book with serial numbers. Each week a report is also made from each point at which wheels are scrapped, giving the number and the cause; and when these are also entered, we have a complete record of dates and places of the "birth, marriage, and death" of each wheel.

This record is a most interesting one, as it gives the service which the wheels perform. Samples of the several forms referred to, both for making the reports and recording the information, are presented further on.

In looking over such a record it is at once apparent that a large number of wheels are always damaged by improper handling or usage. The proportion of *slid-flat* wheels has been very large heretofore; we hope that the more general use of air brakes will effect an improvement in this line. Chipped flanges and worn flanges are also not generally chargeable to the wheels themselves.

What most interest the founder are those which are scrapped for *cracked plates*, *cracked brackets*, and *broken-to-pieces*, as these show improper manufacture in the majority of cases. As the Master Car Builders' regular guarantee is for forty-eight months, all wheels which run over five years may be considered as giving good service; and if these wheels can be obtained, the study of their analysis is very beneficial. Therefore, wheels giving a very short service or a very long one should be carefully inspected and analyzed, as they will show the bad and good features in a much more forcible way than drop or thermal tests. Mr. W. W. Davis, chemist of the Norfolk & Western Railway, who with the writer has given much attention to this

subject, analyzed a large number of wheels giving a long or short service or tested by the drop or thermal methods, and we present the results of some of his work :

	Analyses of Car Wheels.					
	Which Stood Thermal Test for 60 Mins.		Which Stood 40 or More Blows Drop Test.		Which Gave 5 or More Years of Service.	
	Max.	Min.	Max.	Min.	Max.	Min.
Graphite	3.28	2.65	3.31	2.65	3.18	2.23
Combined carbon	.95	.32	.90	.55	1.24	.56
Silicon	.75	.50	.70	.50	.94	.58
Manganese	.53	.20	.46	.24	.34	.13
Sulphur	.088	.055	.086	.040	.085	.047
Phosphorus	.48	.35	.52	.36	.49	.25

It will be seen that these limits are rather wide, but below are given what are considered to be the desirable limits for the chemical constituents of wheels:

Desirable Wheel Analysis.

Graphite	2.75	per cent.	to	3.00	per cent.
Combined carbon	.50	"	"	.75	"
Silicon	.50	"	"	.70	"
Manganese	.30	"	"	.50	"
Sulphur	.05	"	"	.07	"
Phosphorus	.35	"	"	.45	"

By comparing the above with the analyses of a large number of wheels it was found that these limits excluded those which broke through rim in ten minutes or less under the thermal test; broke with twenty blows or less under the drop test; or gave less than two years' service. This last is important, as a wheel can easily be made to stand strains, but the wear will be unsatisfactory, and the metal must have enough chill to stand the abrasion of the track. It is possible to obtain a chill three-quarters of an inch deep on the tread with these proportions.

The analyses of seven wheels which had given from eight to eleven years service each had the following limits :

Graphite	2.56	per cent.	to	3.10	per cent.
Combined carbon	.63	"	"	1.01	"
Silicon	.58	"	"	.68	"
Manganese	.15	"	"	.27	"
Sulphur	.05	"	"	.08	"
Phosphorus	.25	"	"	.45	"

Fig. 216 shows the comparison between chemical analyses of pig iron and of the test bar made by remelting the pig in a cupola by itself. Twenty-seven different irons were examined

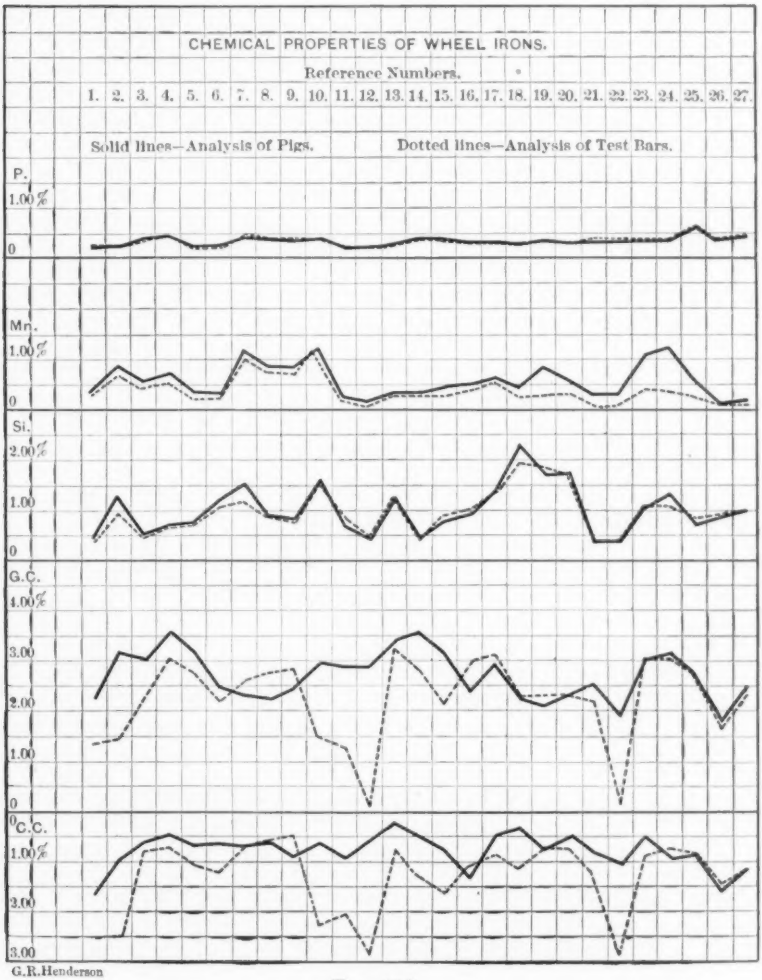


Fig. 216.

in this way. The solid line represents the analysis of the pig and the broken line that of the test bar. It will be seen that phosphorus remains constant, as does silicon when present in quantities under one per cent., whereas some of the manganese

always disappears. The total carbon remains the same, but the proportions of graphitic and combined carbon seem to vary in an erratic manner. This is evidently due to the effect of the chill, as the analyses were made from the chilled test bars when possible, and the great increase in combined carbon in tests Nos. 11, 12, and 22 may be accounted for by the fact that they chilled clear through the two inches.

From the above it will appear that the metal charged into the cupola should contain more graphite, silicon, and manganese and less combined carbon than are desired in the finished product.

If the iron in stock is deficient in manganese, the proper amount can easily be added to the mixture in the ladle by the use of ferro-manganese containing about 80 per cent. manganese, 13 per cent. iron, and 6 per cent. carbon (combined). The use of large proportions of scrap makes this particularly useful, and it is of great assistance in enabling the wheel to stand the thermal test and consequently the strains from severe braking when in service.

Indeed, cracked plates are almost a thing of the past with foundries keeping the manganese up to the limits prescribed above, where previously much trouble was due to that cause. To overcome this it appears that one-quarter of one per cent. should be present in the wheel. Ferro-manganese, however, seems to reduce the chill by increasing the graphitic carbon, as Fig. 217 will show, and this must be allowed for in proportioning the charge for the cupola. The combined carbon is partly changed to graphite, but not entirely in proportion to the amount of ferro-manganese used.

The experiments from which results Fig. 217 was constructed were made by filling small ladles containing different proportions of ferro-manganese from a large ladle which had been filled with the regular wheel mixture, and casting test bars from these small ladles. The effect of this is plainly illustrated both in altering the structure of the iron and the condition of the carbon content.

In preparing specifications for charcoal irons the difficulty of obtaining limits which should be perfectly satisfactory was not underestimated. It is therefore hoped that these will be criticised more as representing an effort to compass a complicated subject than as an absolute specification which has stood the test of time.

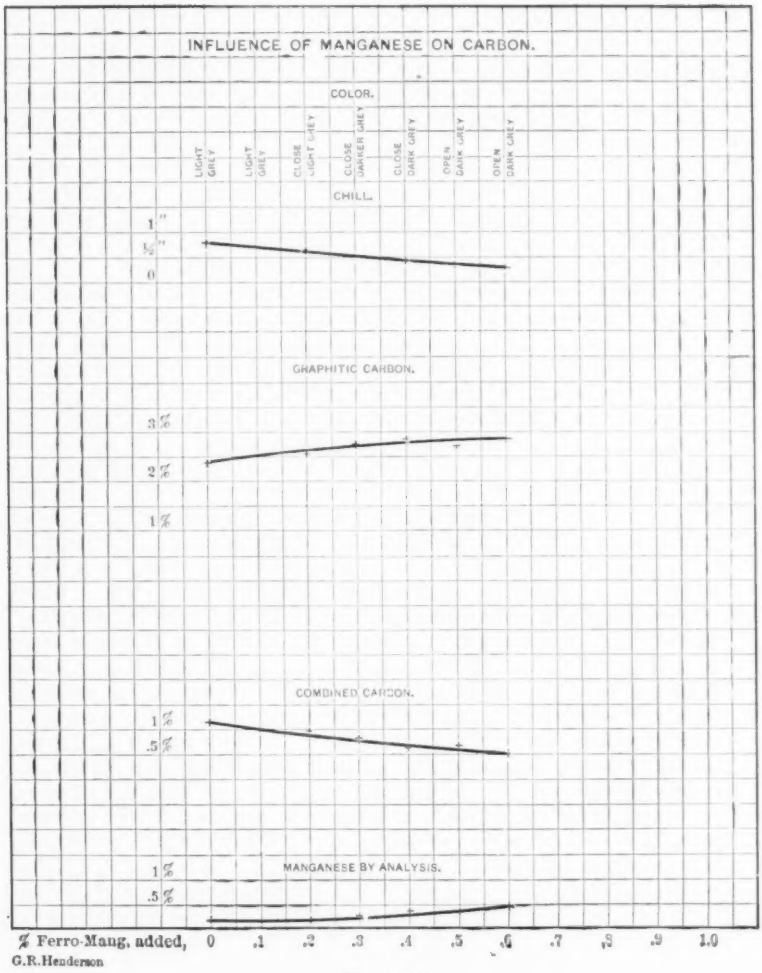


FIG. 217.

Charcoal Pig Iron.

"The material desired under this specification is a charcoal iron with chilling properties as designated below, the various grades to conform to the detail specifications:

Grade.	Chill.	G. C.	C. C.	Si.	Mn.	S.	P.
2	¼ inch	2.50-3.00	.40- .90	1.25-1.75	.50-1.00	.03 Max.	.25-.40
3	½ "	2.25-2.75	.50-1.00	1.00-1.50	.50-1.00	.03 "	.25-.40
4	¾ "	2.00-2.50	.75-1.25	.75-1.25	.50-1.00	.035 "	.25-.40
5	1 "	1.75-2.25	.90-1.40	.50-1.00	.25- .75	.035 "	.25-.40

"The chill is to be measured in a test bar 2 inches square by 24 inches long, the chill piece to be so placed as to form part of one side of the mould. The actual depth of white iron will be measured. The white and gray portions should blend without a definite line of demarcation. The test bars will be made by melting in a small cupola several pigs of the shipment without admixture with other irons.

"The chemical proportions given above are the amounts of the constituents desired in per cents., and will be determined by the analysis of mixed borings in equal quantities taken from 12 pigs from each car.

"In addition to the chilled test bars above mentioned, some unchilled bars of the same size will also be made. These will be placed upon supports 21½ inches apart. A centre load will be applied and the deflection measured. The bar should not break with less than 9,000 pounds, and the deflection at centre with this load should not be less than 0.15 inch."

While charcoal iron is the principal ingredient in car wheels, it is often advantageous to use a small amount of coke iron, and the following specification is therefore submitted :

Coke Pig Iron.

"The material desired under this specification is an open-grain foundry pig conforming to and graded by the following detail specifications :

Grade No.	1,	Silicon	from	3.00	per cent.	to	2.50	per cent.
" "	2,	"	"	2.50	"	"	2.00	"
" "	3,	"	"	2.00	"	"	1.50	"
" "	4,	"	"	1.50	"	"	1.00	"

"Iron will be rejected which shows :

Combined carbon below.....	.10	per cent.	
Graphitic carbon below... ...	3.00	"	
Manganese below.....	.50	per cent.	or above .80 per cent.
Phosphorus below.....	.50	"	" " .80 "
Sulphur above.....	.06	"	

"Should the silicon vary from standard grading, the next lowest grade under the specifications will be paid for, and should the silicon fall two grades low the material will be rejected. No higher grade is desired and no excess price will be paid for higher silicon.

"Samples of twelve pigs will be taken from each car, two from

the top of each end, two from the middle, and two from the bottom. The borings will be mixed in equal quantities and the resultant sample represent the shipment."

Reference having been made to the drop and thermal tests, it may be well to conclude this paper with a specification for car wheels, which will give the details of these tests for ready reference.

33-inch Cast-Iron Wheels.

"The design of wheels must be such that they will be in accordance with the measurements shown on the drawings for 33-inch wheels, and also such that the wheels when cast shall weigh between 590 and 610 pounds. The tread and flange are to be identical with the M. C. B. standard section for same, and the flange thickness must pass the M. C. B. gauges for same.

"Wheels must all be cast in true metallic chills of the same internal diameter and of uniform cross-section. The body of wheel must be of clean, soft, gray iron, and smooth, and free from slag and blowholes, and hubs must be solid and free from drawing. The tread and throat of wheel must be smooth, free from deep and irregular wrinkles, slag, and sand-wash, and practically free from chill cracks and sweat. The depth of clear white iron must not exceed $\frac{7}{8}$ inch at throat and 1 inch at middle of tread, nor be less than $\frac{3}{8}$ inch at throat and $\frac{1}{2}$ inch at middle of tread. Neither should there be a variation throughout the same wheel of more than $\frac{1}{4}$ inch in depth of chill. The blending of the white iron with the gray iron behind it must be without any distinct line of demarcation.

"Each wheel must be so nearly circular that when a true metallic ring is placed upon the tread, and bears somewhere on the cone, it shall at no part of the circumference stand more than $\frac{1}{16}$ inch from the wheel tread. No wheel made in a solid chill will be accepted whose circumference differs more than $1\frac{5}{8}$ inches or less than $\frac{7}{8}$ inch from the circumference of the chill in which it is made. Wheels cast in contracting chills should not differ in circumference more than 2 inches from that of the chill.

"All wheels must, during inspection, receive three heavy blows with a 6-pound sledge at as many different points under the flange, between the brackets, without cracking flange or brackets.

"Each wheel must also be capable of standing a pressure of 50 tons when being mounted on the axle.

"For each pouring of 100 wheels, or fraction thereof, two addi-

tional wheels must be furnished for test purposes. These wheels shall be selected by the inspector, and subjected to the following test: One wheel shall be placed flange downward on an anvil block weighing not less than 1,700 pounds, set on rubble masonry two feet deep, and having three supports not more than 5 inches wide for the flange of wheel to rest upon; it shall be struck centrally upon the hub by a weight of 140 pounds falling from a height of 12 feet. Should this wheel stand 15 blows without breaking in two or more pieces, the lot of 100 may be accepted (as far as this test is concerned), provided that the broken wheel shows proper depth and uniformity of chill; also subject to return if the wheels do not satisfactorily stand boring and mounting, as explained above. Should the wheel break in two or more pieces with fifteen or less blows, the lot of wheels represented by the one tested will be rejected.

"The other test wheel must be laid flange down in the sand, and a channelway, 1½ inches wide and 4 inches deep, must be moulded with green sand around the wheel. The clean tread of the wheel must form one side of this channelway, and the clean flange must form as much of the bottom as its width will cover. The channelway must then be filled to the top with molten cast iron, which must be hot enough when poured so that the ring which is formed when the metal is cold shall be solid or free from wrinkles or layers. The time when pouring ceases must be noted, and two minutes later an examination of the wheel must be made.

"If the wheel is found broken in pieces, or if any crack in the plate extends through the tread, the 100 wheels represented by the test will be rejected.

"Wheels may be dried before submitting them to the thermal test, but must not be warmer than 100 degrees Fahr."

DAILY REPORT OF WHEELS MADE.

On _____ 18

Wheels were made at _____ Foundry as follows:

Wheel Numbers.	Diameter.	Kind of Chill.	Mixture.		
			Iron.	No.	Per Cent.

[Eleven lines.]

Wheels Spoiled.

_____ 18 _____ Foreman.

Report of Wheels Mounted.

At _____ Shop during the week ending Saturday _____ 18

Number.	Number.	Number.	Number.	Number.	Number.	Number.	Number.

[Sixteen lines.]

Report of Wheels Scrapped.

At _____ during the week ending Saturday _____ 18

Wheel No.	Defect.	Wheel No.	Defect.	Wheel No.	Defect.

[Fourteen lines.]

Classify defects as follows: Shelled Out, Slid Flat, Seams, Worn Out, Worn Flange, Worn Hollow, Burst, Chipped Flange, Chipped Rim, Cracked Tread, Cracked Plate, Cracked Brackets, Broken to Pieces.

_____ 18 _____

Wheel Drop Test Report.

Date Cast.	Wheel Number.	Shrinkage.	Depth of Chill.				Blows to		Remarks.
			Throat.		Tread.				
			Max. 7-8″.	Min. 3-8″.	Max. 1″.	Min. ½″.	Crack	Break.	

[Fourteen lines.]

_____ 18 _____ Foreman.

Daily Report of Thermal Test of Wheels.

On _____, 1898, wheels were tested as below:

Date Cast.	Wheel No.	Cracked.			Chill.		Remarks.
		Bracket.	Plate.	Rim.	Throat.	Tread.	

[Ten lines.]

_____ Foreman.

Wheel Record.

Wheel.					Mounted.		Scrapped.			Months' Service.
No.	Diam.	Chill	Place Cast.	Date Cast.	At.	Week Ending.	Place.	Date.	Defect.	

WHEEL DROP TEST AND MIXTURE RECORD.

Date Cast.	Wheel Number.	Shrinkage.	Depth of Chill.				Blows to		Remarks.	Mixture by Per Cents.											Wheel Scrap.
			Throat.		Tread.																
			Max. $\frac{3}{4}$ in.	Min. $\frac{3}{4}$ in.	Max. 1 in.	Min. $\frac{1}{4}$ in.	Crack.	Break.													

DISCUSSION.

Mr. Frank M. Ashley.—I do not find in this paper one of the things which I naturally expected concerning the method used in chilling the surface of the treads.

The practice followed by Mr. Geo. W. Miltimore, of Arlington, Vt., interested me on a recent visit. He places two wheels together, with their faces running in opposite direction, and allows a chill to follow from the friction generated between the two surfaces. A fine file could hardly touch the wheel after it came from the abrasion treatment. I would like to be told how that work is done by other manufacturers.

Mr. William Kent.—I fear that this paper is not likely to have the discussion which it deserves, because it happens that there are few present here well enough acquainted with the subject. I wish that members could be persuaded to send in written discussions, and that this subject might be carried over for discussion at two or three meetings. I believe much good could come from a topical discussion in answer to the question "What do you know about the strength of car-wheel irons?" This paper also ought to interest that committee of the American Foundrymen's Association who are investigating this subject. I believe our greatest advances in the strength of iron are to be found by ollowing the car-wheel iron practice.

*Mr. George R. Henderson.**—Regarding the method of chilling which Mr. Ashley speaks of, I will say that this is novel to me, as I have never seen an operation of this kind. The method of chilling wheels both by solid and contracting chill has been so long in use in wheel foundries that I hardly thought it necessary to go into this detail in the paper itself, as, in fact, car wheels would be useless and impossible to manufacture without the chill in which they are cast.

* Author's closure, under the Rules.

The principal chills used nowadays seem to be the Whitney, Barr and Canda, and while there are differences of construction, yet the principle and the effect are practically the same.

As the author is himself desirous of obtaining the views of members on this interesting question, he would be very glad if Mr. Kent's suggestion of a further discussion of the matter could be carried out.

DCCCXIII.*

THE PLUNGER ELEVATOR.

BY GEO. I. ALDEN, WORCESTER, MASS.

(Member of the Society.)

Introduction.

The plunger elevator as a lift, or freight elevator, is well known. As a passenger elevator for short runs it has been used more or less for some time, both in this country and in Europe. It has always been characterized as a safe, durable, and economical machine, but it has generally been assumed that it had limitations which prevented it from meeting the requirements for passenger service, except for short runs and slow speeds. The suggestion of its use as a high-speed passenger elevator would until recently have been regarded as indicating a serious misconception of its possibilities.

Recent experience, however, seems to indicate that the plunger elevator is a satisfactory solution of one of the most important engineering problems connected with the special and complicated equipment of a modern office building.

The principal parts of the elevator are shown in the accompanying cut. Near the level of the ground the cut shows the casing—which in soft ground is driven down to enclose the cylinder proper—broken away, showing the cylinder. The latter is also broken to show the plunger passing down inside the cylinder. The annular space between the plunger and the inner walls of the cylinder is ample for the passage of the water down around the plunger as the elevator ascends, or up as the elevator descends.

The cylinder above the casing terminates in the cast-iron cylinder head, to which is bolted the stuffing-box for the plunger, and to which also are attached the buffer springs upon which the car rests when in its lowest position.

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* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

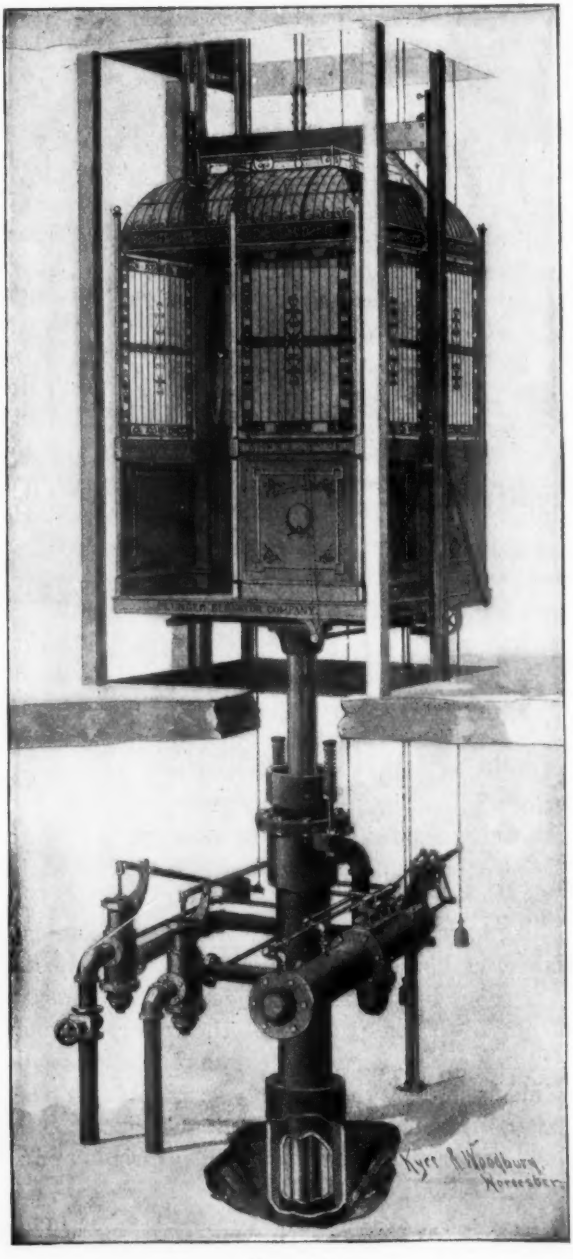


FIG. 218.

The plunger at its upper end enters a steel casting into which it is fastened as shown in Fig. 220. This casting is bolted to a platen on the under side of the car.

The metal cab, of ornamental design and finish, rests upon the car floor or platform. Channel irons pass up from the car floor on two sides, and are connected at their upper ends by cross beams to which the counterweight ropes are secured. These ropes pass up and over the large sheave at the top of the well-room, and carry the counterweight, which runs in guides at the side of the well-room.

Returning now to the basement, the three-way main controlling valve, with its pilot valve, is seen in a horizontal position in front of the cylinder. Water is supplied through the vertical pipe on the left. This is provided with a shut-off valve. At the upper end of this vertical pipe a long-turn elbow leads the water to one of the supplementary valves, from which it passes through a pipe behind the main cylinder to the main controlling valve. A second supplementary valve is shown on the exhaust pipe, which leaves the controlling valve in front and at the left of the cylinder, and disappears through the floor just at the right of the supply pipe.

The cab is shown just above the first floor, in Fig. 218, the enclosure work at the floor above, and the overhead work in Fig. 219.

Cylinder.—The cylinder is sunk plumb into the ground until its upper end is a little below the lowest floor to which the elevator is designed to run. It is a steel tube, having its lower end closed, and its upper end fitted with an enlarged cylinder head, which includes a stuffing-box for the plunger, and an opening to receive the pipe through which water is admitted to and discharged from the cylinder. The bursting pressure for a 10-inch cylinder is about 3,750 pounds per square inch. This is figured from a tensile strength of the material of 60,000 pounds, and a thickness of cylinder walls of $\frac{5}{16}$ inch.

The water pressure commonly used is from 140 to 150 pounds, or in some cases 200 pounds to the square inch.

The sections of which the cylinder is made are squared up and threaded in a lathe, so that when connected by couplings the cylinder is straight. When finished it is tested by hydraulic pressure, and then coated with a good preservative paint.

Drilling.—The well for the cylinder is made plumb, so that the cylinder may stand in a vertical position. If the subsoil is earth, a steel pipe of diameter large enough to receive the cylinder, and

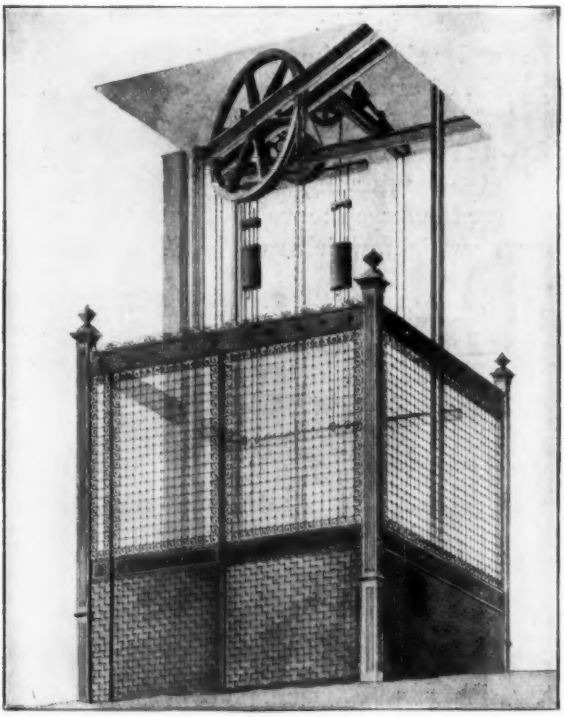


FIG. 219.

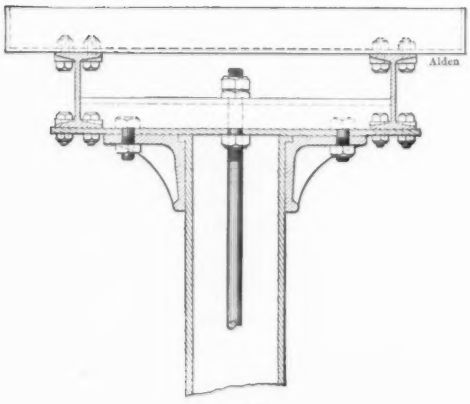


FIG. 220.

called the casing, is first driven to the required depth, the earth being removed from the inside of this casing as it is driven. If ledge is encountered, it is drilled to receive the cylinder, and no casing is required.

Plunger.—The plunger is made of steel tubing which is straightened, turned to uniform size, and polished. The sections are united by a special joint, which is clearly shown in the sectional drawing (Fig. 221). This joint has a strength to resist cross-breaking or buckling fully equal to the strength of the plunger at a cross-section between the joints. One of these joints was recently tested at the Worcester Polytechnic Institute for resistance to cross-breaking. The supports were 74 inches apart, and the joint was 2 inches from the centre, the load being applied at the centre. The moment of inertia of the cross-section of the plunger was 21.3; the load at elastic limit was about 10,200 pounds; deflection, $\frac{18}{100}$ inch; the stress, 28,000 pounds per square inch; the breaking load, 23,500 pounds; and the breaking stress, 60,153 pounds per square inch. In this test the joint began to open a little on the convex side at about the same time that the tube showed signs of weakness near the end of the coupling nipple. The bulging of the tube near the end of the coupling nipple indicates that if bending had been carried to a greater extent the short bend (for the pipe would not break if bent double) would have occurred here, and the joint would not have failed.

The diagram showing loads and deflections for the test piece above referred to is given in Fig. 223.

The top end of the plunger is flanged into a steel casting, which is securely bolted to a steel platen, which in turn is securely bolted to the car. In long-run elevators a rod or, better, a loop of galvanized wire rope passing inside the plunger has an independent connection with the car, which would securely attach the weight of the plunger to the car if the regular fastening should fail.

The strength of a plunger to sustain the car and its load is a matter which it is of prime importance to consider. The natural first impression that a steel column 6 or 8 inches in diameter and 150 to 200 feet long is very slender and will be in great danger of buckling, even under a very small load, needs to be corrected.

A consideration of the formulas, together with the conditions under which the plunger is used, and the facts which have been

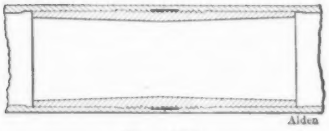


FIG. 221.

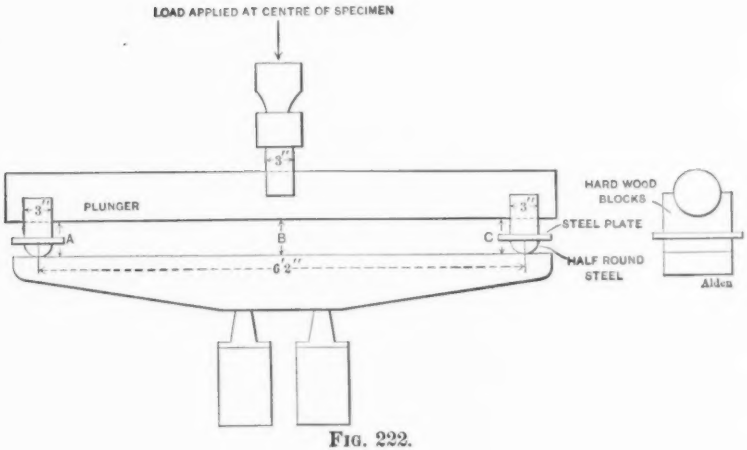


FIG. 222.

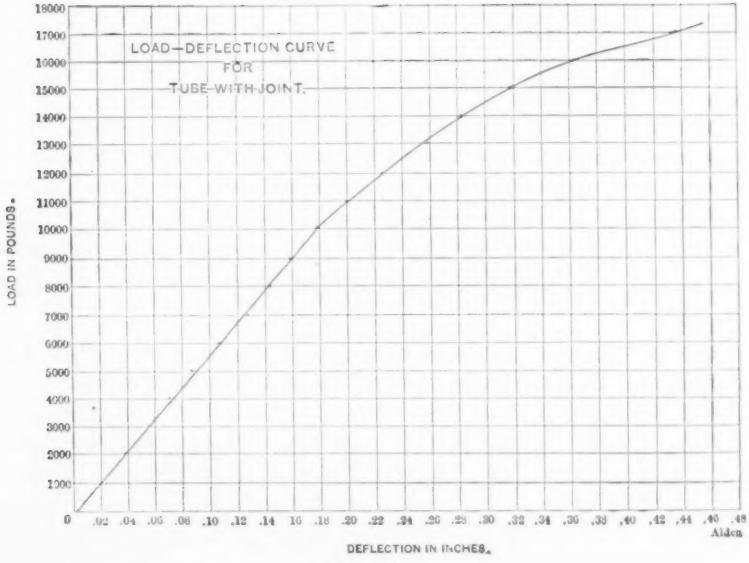


FIG. 223.

established by experiment and experience, ought to lead us to safe conclusions on this point.

For a long column with rounded ends the ultimate load is given by the formula

$$P = \frac{\pi^2 EI}{L^2}, \quad \ldots \ldots \quad (1)$$

in which E is the modulus of elasticity of the material; I the moment of inertia of the cross-section, and L the length of the column.

If the column has flat ends, or is fixed in direction at the ends, the ultimate load is

$$P = \frac{4\pi^2 EI}{L^2} \ldots \ldots \quad (2)$$

The case of an elevator plunger comes somewhere between these two extremes. In an experiment tried at the Washburn Shops of the Worcester Polytechnic Institute in 1894, when an elevator having a 4-inch plunger was loaded up to its ultimate load, it was found that this load was given approximately by the formula

$$P = 2.3\frac{\pi^2 EI}{L^2} \ldots \ldots \quad (3)$$

Formula (1) applies to any upright strut or pillar whose length is so great in comparison with its diameter that the only stress that needs to be considered is that due to bending. In such a pillar the first indication of the approach to the maximum load is the bending sideways of the pillar and the increase of this bending by slight increments of load. When the bending is well established, the pillar is theoretically in a state of indifferent equilibrium; that is, the same load will hold the pillar to its curved form, whether bending be much or little, so long as the bending is not sufficient to cause a moment on the cross-section exceeding the proof moment of stress of the section. A pillar thus loaded with its ultimate load might continue to bend indefinitely and finally break, by the addition of a small increment to the load or by any slight external force applied to increase the bending.

As an elevator plunger is in a condition somewhere between a pillar rounded at the ends and a pillar fixed in direction at the ends, the action of the plunger under an excessive load will be

different from that of a pillar rounded at the ends, as just described, in this respect, viz.: instead of the plunger being in a state of indifferent equilibrium after bending is established, it will be stable, for the reason that the car at the top, and the stuffing-box at the bottom, offer an increasing resistance to the change of direction of the plunger at its ends. Therefore, after a plunger is overloaded so that it begins to bend considerably, it will require quite an increment of load to much increase the bending.

For example, a plunger 42 feet long and 3.92 inches outside diameter showed a deflection from the vertical as follows:

Net Loads.		Deflections.	
5,000	pounds	$1\frac{7}{16}$	inches
6,000	"	$1\frac{7}{16}$	"
7,550	"	$2\frac{1}{16}$	"
8,500	"	$3\frac{3}{4}$	"
8,900	"	$4\frac{31}{32}$	"

It is well to have the above analysis in mind when considering the question of factors of safety. To follow here the general rule that there should be a factor of from 4 to 10, and then to conclude that as an elevator should be abundantly safe it would be well to use a large factor, would be allowing the general rule to take the place of reason. In the case of cross-breaking, if there is a factor of safety of 4 or 5, as in bridgework, for example, every time the maximum working load is applied it produces a stress equal to one-fourth or one-fifth of the breaking stress, and it would be unsafe to use a smaller factor. But in a long upright pillar, as used for an elevator plunger, the case is entirely different. A load nearly equal to the ultimate load may be applied an indefinite number of times, and not cause a stress that would be even worth considering, because it is only when bending has begun, and been carried to a considerable extent, that any severe stress is brought upon the metal.

If, therefore, we find by formula (1) that for some given case $P=8{,}000$ it is perfectly safe to use a steady working load of 5,000 pounds, or 6,000 pounds as the maximum working load. The only thing to be avoided is the load that will produce bending, because up to the time that serious bending occurs the metal is under no stress except a light thrust. In the case of a long plunger in an elevator well, the effect of accidental unreasonable overloading would be to bend the plunger until its convex side should reach the side of the elevator well or the edge of one of

the floors of the building. It would then be impossible to bend it more, and the car could not fall, even if loaded far beyond the ultimate load given by the formula.

This is indicated theoretically by solving the following problem: Given, a plunger $6\frac{1}{2}$ inches outside diameter, 6 inches inside diameter, and 150 feet long; required, the deflection at or near the elastic limit.

If this plunger were simply supported or held in place at the ends, and a force applied at the centre to deflect it sideways, it would require about 460 pounds to cause a stress due to bending of 28,000 pounds per square inch. The deflection from a vertical line which this force would cause is about 77.6 inches, or nearly $6\frac{1}{2}$ feet. The limiting load for this plunger would be by formula (3) about 5,000 pounds, assuming that $E=30{,}000{,}000$ pounds per square inch and $I=24$, and neglecting the weight of the plunger itself so far as this weight would affect deflection.

Let us now examine the plunger as regards stress, when the elevator is in actual use and carrying a maximum live load of 3,000 pounds, or about twenty people of average weight.

The plunger itself would weigh about 2,400 pounds; the car might weigh about 3,000 pounds; the compensating cables about 1,100 pounds; the counterweight, say, about 3,500 pounds. With a water pressure of 150 pounds per square inch at the starting level, the pressure on the lower end of the plunger when the car is at the top of its run is about 5,000 pounds, and as the plunger weighs 2,400 pounds there could be but 2,600 pounds pressure, or load on the upper end of the plunger, even when the load is being accelerated along its upward run. When the elevator is standing at the top of its run with its maximum live load, 3,000 pounds, there is a load of 1,400 pounds on the plunger; and when standing with no live load, there is no load on the upper end of the plunger, but the plunger is under tension from the top down to a neutral point 50 feet from its lower end. When the elevator is descending with its full load and being brought to rest, the thrust on the plunger would be increased somewhat above 2,600 pounds.

In the above example, which is used simply as an illustration of the action and effect of loads on the plunger, the friction of the plunger in its stuffing-box, the overhead shaft, and the guide-shoes, and the resistance due to stiffness of ropes were not considered. The complete formulas for expressing the relations of

the various quantities entering into the consideration of lifting capacity are as follows:

Let W = Weight of plunger.
Let C = Weight of car.
Let L = Live load.
Let K = Counterweight.
Let F = Resistance due to friction.
Let S = Sectional area of plunger.
Let p_0 = Water pressure per square inch at top of cylinder.
Let R = Weight of compensating ropes per lineal foot.
Let T = Total run of elevator.
Let A = Unbalanced force to give upward acceleration with live load.
Let A_1 = Unbalanced force to give downward acceleration without live load.

As the compensating cables make the lifting capacity constant throughout the run, or in other words counterbalance the varying pressure on the plunger due to change of elevation of the car, this variation need not be taken into account in the formula.

Therefore the equations may be made considering the car at the top of its run. Then

$$A = p_0S + K + RT - W - C - L - F. \quad . \quad . \quad (4)$$

The equation for A_1 should be for the elevator without its live load, thus

$$A_1 = W + C - K - RT - F. \quad . \quad . \quad . \quad (5)$$

In these equations p_0 and K are the arbitrary constants whose value is to be fixed so as to secure the required speed of the loaded car up and of the unloaded car down. The value of K that makes $A = A_1$, found by combining (4) and (5), is

$$K = W + C - RT - \frac{p_0S - L}{2}. \quad . \quad . \quad . \quad . \quad . \quad (6)$$

In practice it would probably be well to make A a little greater than A_1, as there is less mass to be accelerated when the empty car is coming down than when the loaded car is going up.

If, with an assumed value of p_0, K, as found from (6) and substituted in (4) and (5), does not make A or A_1 large enough to give the required acceleration, p_0 may be increased; that is, a higher pressure may be carried in the pressure tank.

Valves.—The main controlling valve, which is a three-way balanced valve, is usually moved by a motor piston, which is in turn operated by a pilot valve. This pilot valve is controlled by a lever in the car. The supplementary valves close automatically and bring the elevator to an easy stop, both at the top and at the bottom of the run.

The overhead work consists of a large sheave for the counterweight ropes, and smaller ones for the ropes that operate the supplementary valves. These sheaves are mounted on heavy steel shafting supported in bearings with drip-pans; the bearings rest on iron beams securely built into the walls of the building.

Power.—The power for running the elevators is usually furnished by a pumping plant. In some cases water from the city mains is used, but where the elevators run frequently it is generally cheaper and better to pump the water and use it over and over. The pumping plant may be for steam or electric power. It consists of a closed steel tank which is kept about $\frac{6}{10}$ full of water and $\frac{4}{10}$ full of air. The capacity of this tank is from ten to twenty times the volume of water used per trip by one elevator. The pressure in this tank is usually maintained at about 150 pounds per square inch. From this tank water is drawn to send the elevator up. There is also an open tank of somewhat less capacity than the closed one, which receives the water that is discharged from the elevator cylinder as the elevator descends, and a pump which draws its supply from the open tank and delivers to the closed tank. A relay pump is also usually provided.

The pump may be a steam pump driven from high-pressure boilers and exhausting its steam during the cold weather into the heating system, in case the building is heated with low-pressure steam; or it may be a power pump driven by an electric motor or other convenient means. In either case the pump is supplied with an automatic regulator, which stops the pump when the pressure in the closed tank reaches its maximum, and starts it when the pressure falls a few pounds below this maximum.

Operation.—When the plant is completed and ready to start the pressure tank is charged with air and water. Air is admitted through the suction pipe of the pump and forced into the tank along with the water. To occasionally supply any loss of air from slight leakage or other cause, a device, which is in principle a small air pump using water as a piston, is introduced between the water valves of the pump and the delivery pipe.

In a steam pumping plant a regulator in the pipe supplying steam to the pump is so adjusted that when the pressure in the tank reaches its maximum the regulator shuts off the steam, and this stops the pump. When the pressure in the tank falls a few pounds below the maximum, the steam is let on and the pump starts and runs until the maximum pressure is again restored.

In an electric pumping plant, in place of the steam-regulating valve, an automatic switch, operated by pressure from the tank, stops and starts the motor, as the pressure in the tank rises or falls.

The tank being properly charged, the air expelled from the cylinder and piping connecting it with the tank, and the elevator valves being adjusted, the elevator may be started. We will assume that it is a passenger elevator having a pilot valve, and a lever in the car for operating it. The lever being in its central position, the main valve is closed. The lever is held in this central position by a catch which is easily released by the pressure of the hand on the handle of the lever. The lever is then thrown a little to one side. This opens the pilot valve and allows water from the pressure tank to enter the motor cylinder and move the motor pistons. The motion of the motor pistons moves the main controlling valve, and it begins to open; but this motion of the main controlling valve closes the pilot valve, so that if the lever is thrown but a little from the centre, the controlling valve will open but a little and the elevator will move slowly; throwing the lever farther from the centre causes the controlling valve to open still wider and increases the speed of the elevator.

The ports in the valve being properly proportioned with reference to the motions of the valve, the elevator is under the most complete control of the operator. The valve is adjusted so that if the lever is thrown quickly to the centre when the elevator is going up at full speed the plunger will not leave the water, but the elevator will make a gradual and gentle stop. The action of the valve mechanism is so perfect that when the elevator is descending at full speed the lever may be thrown quickly from one side to the other. This will bring the elevator to a stop and start it in the opposite direction without any jar or any sensation to the occupants of the elevator that would call attention to the change of motion from down to up. The perfect control of the elevator, the uniform though rapid acceleration in starting and stopping, and the absence of any vibration from the stretching of ropes are qualities which give to the passengers a sense of security

and prevent the unpleasant sensations which are often experienced from sudden starts and from vertical vibration. As the elevator approaches the end of the run, if the operator fails to throw the lever to the centre and thus close the main valve, the appropriate supplementary valve closes automatically and brings the elevator to rest at the upper or lower floor, as the case may be.

Speed.—High speed in elevator service becomes desirable as the distance the elevator has to run increases. This is particularly true where some of the elevators in a high building are run express. For many years the plunger elevator was built for slow speeds. It had small water pipes and small valve ports. To get high speeds it was only necessary to make large water passages. But the problem of *getting* speed was much simpler than that of securing *perfect control* with high speed. The two important ends to be attained by this control are, first, stopping the car exactly at the floor, and, second, starting and stopping with such an acceleration as to give no unpleasant sensations to the passengers. The first of these ends is attained in the plunger elevator by the use of the pilot valve. When the operator has had a litttle practice in handling the lever, he has no trouble in stopping the car properly.

The plunger elevator apparently presents peculiar obstacles to high speed. The plunger and counterweights are quite heavy, so that in starting there is a large mass to accelerate. If the water could be shut off too suddenly on the upward run, the mass would move on and the plunger leave the water; and on the downward run, too quick a stop would cause a disagreeable pound or thud as the elevator came to rest.

The first of these difficulties (the starting of large masses) is overcome by means of the surplus energy stored in the pressure tank. This tank is charged by the pump, as has been described, and the pump runs much more of the time than the elevator draws water from the tank, and always delivers against about the same pressure. The expenditure of stored energy to accelerate the elevator and its load is therefore not accompanied by any increased stress on the pump, or indeed on any part of the machine. Starting the elevator and its load up causes no sudden pull on ropes, and no periodical increase of load on the overhead work or on building which supports it. To remedy the second seeming difficulty (that of the plunger leaving the water) the first step was to ascertain the maximum distance that the ele-

vator, running up at its greatest speed, would move before coming to rest, if left to itself. In making this calculation, friction and other similar resistances are left out of account, as they help to stop the elevator and therefore make the results on the side of safety.

Let V = Maximum velocity of elevator in feet per second.

Let f = Acceleration of the system when free from any external force.

Let D = Distance passed over while coming to rest.

Let W = Weight of plunger.

Let C = Weight of car.

Let K = Counterweight.

Let R = Weight of compensating ropes per foot.

Let T = Total run of elevator in feet.

Let g = Acceleration of gravity in feet per second.

Let the elevator be near the top of the run and unloaded, as this is the most unfavorable for a quick stop. Then

$$D = \frac{V^2}{2f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and

$$f = -\frac{K + RT - W - C}{K + RT + W + C} g \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Having determined the value of D from (7), the valve should begin to close when the car is at this distance, D, from the point of stopping, and then the closing of the ports should occur at such a rate that the water will just keep the volume generated by a cross-section of the plunger completely filled.

This end is attained by correctly proportioning the valve ports and governing the travel of the controlling valve. A careful study of the problem as above outlined led to the conclusion, which is borne out in practice, that when the problem is solved and the seemingly unfavorable conditions are met, the result is an ideal stop. The elevator stops just as a body thrown into the air comes to rest, so gently that one in the car can hardly tell just when the stop occurs. Thus the conditions which seemed unfavorable or possibly insurmountable have compelled the treatment that is essential to the best results.

Plunger elevators in use at the present time are running at a speed as high as 600 feet per minute.

Economy of Room.—One feature that is of some importance in large buildings is the compactness of the plunger elevator plant, thus economizing room. The cylinder, being in the ground, occupies no room in the building, and the operating valves can often be placed in the well-room sub-basement directly under the car, and even when placed just beside the well-room take up but little space.

Cost of Power.—A steam pumping plant is an economical source of power for running a system of plunger elevators, particularly when the building is heated by steam. The water pressure being applied directly to the load through the plunger, the elevator requires a minimum quantity of water per trip. The expense for coal can be figured as follows: Determine the probable average number of trips the elevator will make daily. Compute the number of cubic feet of water required to make this number of trips, and then the number of foot-pounds of energy required to pump this volume of water against the pressure to be maintained in the pressure tank; divide this number of foot-pounds by the duty of a steam pump, and you have the number of hundred pounds of coal required. From the cost of coal may be deducted the value of the steam that can be used for heating, the balance remaining being the net cost of fuel to run the elevators.

Repairs.—The cost of repairs on a plunger elevator for a series of years is very small. No essential part of the mechanism is under any severe strain or subject to any destructive wear. The life of a plunger elevator is therefore very long. There are elevators running on city water that have been in constant use for over twenty-five years. Plunger elevators have not yet been made long enough to determine the real limit of their useful life, but have been made long enough to show that this period is a very extended one. Packings for the plunger and valves, and guide shoes for the car, are about the only parts that require renewal.

Safety.—The elements of safety in the plunger elevator may be summarized as follows: The elevator car cannot fall, first, because the plunger always rests upon a column of water which can escape only as it is driven out of the cylinder through comparatively small openings; and, second, because the plunger is of steel, that would bend to the side of the well-room without being put under dangerous stress. The elevator car cannot be made to shoot up

to the top of the well-room, first, because the elevator as a whole can only be carried up by the pressure of water which enters the cylinder through the small valve openings; and second, because the plunger is so securely fastened to the car by two separate and independent means that it cannot become detached, which is the only way in which the car can be drawn up by the counterweights. In case of simultaneous failure of both the main controlling valve and the supplementary valve to work at the top of the run, the counterweight would reach the ground, and the car would stop.

In case of the failure of both the main and supplementary valves at the bottom of the run, the car would land upon the buffer springs, and would stop with somewhat of a jolt, which would be disagreeable to passengers, but ought not to be serious. If the elevator were running 600 feet per minute, or 10 feet per second, the effect of a sudden stop would be the same as if the elevator should fall freely from a state of rest through about 19 inches. If the speed were 7 feet per second, which is even greater than that of most elevators, the effect of a sudden stop would be the same as if the elevator should fall freely about 9 inches.

Finally, to be sure that the elevator will never attain a downward speed greater than the maximum predetermined for that particular elevator, a special throttle valve is introduced into the exhaust pipe. This valve is lifted to its seat by the pressure due to the velocity of the exhaust water, and thus it works only when the elevator has attained a certain speed. When seated, it throttles the exhaust so as to reduce the speed of the elevator considerably below its predetermined maximum, and this lower speed cannot be exceeded until the elevator has been stopped. As soon, however, as the elevator is stopped, the throttle valve drops from its seat and leaves the exhaust passage again free.

This valve is a perfect check on the speed of the elevator, and is entirely out of the reach of the elevator boy. When once adjusted it absolutely prevents the acquirement of excessive speed either from carelessness or from loss of control of the operating valve, and when the maximum speed is attained this valve acts instantly and reduces the speed considerably below that maximum.

If an analysis of the fatal accidents due to failure of elevators should be made, and the cause of each accident determined, and if it were considered what would have happened if, with no other change in conditions, a plunger elevator, as at present constructed, had been in use at the time of the accident, I venture to

suggest that it would be difficult to escape the conclusion that the accident would, in nearly every case, have been avoided.

Conclusion.—The modern office building must, from its height and weight, be constructed in a very thorough and substantial manner. Often it is architecturally beautiful, and no expense is spared to secure the best material and most elaborate ornamentation. It is built to attract tenants, and to compete with other buildings in giving its occupants comfort, safety, and convenience. Nothing is more essential to the physical safety and the mental comfort of those who are obliged to use the elevators several times a day than that these elevators should be the safest it is possible to make, and that they should also be of such substantial appearance and such smooth and quiet motion as inevitably to impress the passenger with a sense of security. Too often the elevators are a secondary consideration. Money is spent lavishly on costly arches, pillars, staircases, and various showy architectural features, and when it comes to the means of safe and quick access to the offices and stores, there is thought to be only money enough left to pay for the cheapest elevators. This practice is, however, being slowly abandoned, for two reasons: first, it proves in a few years to have been the most costly method; and, second, tenants are becoming more intelligent regarding the relative merits of different elevators, and more strenuous in their demands for the best. The successful renting of large office buildings of apartment houses is aided by having a type of elevator that has a good record.

The plunger elevator has, by request, been somewhat fully described in this paper, that the Society may know of its recent development, and may consider its merits as a worthy competitor for first place among the appliances of its kind in a modern building.

DISCUSSION.

Mr. Wm. Barnet Le Van.—I would like to state that I built and erected a hydraulic lift at French, Richards & Co.'s, Tenth and Market streets, Philadelphia, about twenty-five years ago, the ram or plunger of which was 8 inches in diameter and 90 feet in length. The machine was tested for one week, with an average load of 6,000 pounds; the speed of ram was 80 feet per minute, in rising, and when lowering, 160 feet. The coal consumption during the trial averaged 1,000 pounds per day of

eleven hours. The building was 75 × 120 feet; the first floor was used as a salesroom and for offices; the whole of the building was heated from the same fuel used for running the elevator. It was in constant use for seventeen years, during which time the actual cost for repairs was about $150.

The ram and platform were not balanced; it took 300 pounds per square inch water pressure to balance them when the platform was empty.

Mr. William H. Bryan.—Professor Alden's paper raises some interesting questions, the more important of which are how many plunger elevators are actually in service, and to what extremes of lift and speed have they been carried. Is it safe to place 100 feet or more of cylinder in the ground, where it is absolutely inaccessible, and subject to deterioration?

I suggest that Professor Alden explain a little more fully the compensating cable mentioned on page 638.

Professor Alden seems to prove his point of extreme safety, but it is doubtful whether the general public would accept his statement, when, with maximum load and a lift of 150 feet, the deviation of the plunger from the vertical might be noticeable to the eye.

The point of economy of space is a good one, as is also the feature of higher efficiency due to direct connection and absence of sheaves. It, of course, retains the inherent defects of all ordinary forms of hydraulic elevators:

1. That full power is required for all loads, whether minima or maxima.

2. That efficient counterbalancing is impossible, and there is no return of power on the return trip.

3. That the ordinary pumping plant is very wasteful in fuel, while the more efficient types are so expensive as not to be justified except in the case of very large plants.

4. That the power-generating plant of the elevator must be independent of that for lighting, and both must have their independent reserve units, which features increase the cost, complicate the arrangement, and reduce fuel efficiency.

Mr. Boyer.—I think it would be well if we knew the method by which Professor Alden counterbalances that shaft as it goes up. You could imagine starting a cylinder and rising 100 or 125 feet; you are putting a great deal of metal constantly in the air. It might be a pleasure to some of you to know how he counterbalances that.

Mr. Alden.—Regarding what has been done, I will say that at the Bannigan Building, in Providence, there are five elevators running 120 or 130 feet, at a speed of 400 feet or more. At the Jewellers' Building in Boston they run 132 feet. The speed guarantee is 600 feet a minute, and I think that is excelled. The plungers are 6½ inches in diameter. So far as the deflection is concerned, it is not noticed at all. You will find a little tendency in the plunger to get a quarter of an inch out in one direction or another, but that is an initial condition, and that does not usually change. In the Bannigan Building elevators, one is required to carry a weight, 8,000 pounds, to the top of the building. We consequently tested the elevator with a load of 8,000 pounds, and we found that the plunger carried it with perfect safety to the top of the building. I have never known a long plunger in which, with the maximum regular load, there was any deviation from the vertical which could be detected or was noticeable to the eye. Regarding any objection that may be felt against placing the cylinder in the ground it should be remembered that there are no machine parts or pipe connections below the ground; that the cylinder is well protected; and it has been found to be long-lived. During twenty years' experience a cylinder has never been removed for repairs. The cylinder, being in a vertical position at the bottom of an open well, is conveniently removed if occasion should require it.

Regarding the defects which Mr. Bryan has enumerated, the first and second may yet be overcome, some steps having already been taken in this direction. The third is somewhat modified in case of a steam pumping plant in which the exhaust steam is utilized for heating purposes. So far as actual cost of running is concerned, where a steam boiler is maintained for heating, plunger elevators run by a steam pumping plant are probably the most economical system in use.

Of the fourth defect mentioned, it may be said that the requirements of elevator service are such that in case of electric elevators it is doubtful whether they should be run in connection with a lighting system, the sudden loads being detrimental to the steadiness of the electric lights. If storage batteries are used, as has been found necessary in some cases, the additional first cost and subsequent charge for maintenance are very great.

Regarding counterbalancing, it is clear that, as the elevator moves, the pressure on the lower end of the plunger changes.

The amount of this change of pressure for every foot the plunger moves is just equal to the weight of water displaced by the plunger in its motion. As the elevator moves up, its lifting capacity is thus diminished. To counterbalance this change in lifting capacity, we suspend the counterweight by cables of such size and number that the weight of the cables per foot shall be just one-half the weight of the column of water displaced by the plunger. Then as one foot of cable is transferred to one side or the other of the counterweight shaft it just balances that difference, and the lifting capacity is uniform throughout. The strength of the counterweight cables is thus made very much in excess of what is needed to hold the counterweight. Consequently the cables do not need renewal, as they are strained but very little. There is no place in this whole elevator system where there is any extraordinary stress upon any part of the machine, either tensile or thrust, or excessive pressure on wearing surfaces. This condition in a machine so simple in construction accounts for its being very long-lived.

DCCCXIV.*

INVESTIGATIONS OF BOILER EXPLOSIONS.

BY GUS. C. HENNING, NEW YORK CITY.
(Member of the Society.)

In spite of the numerous boiler explosions constantly occurring in all kinds of service, and the numerous investigations of their causes which have been made officially and otherwise, no general method has thus far been proposed which gives any certainty of reliability of conclusions.

The well-known boiler tests made by a government board of the United States, at Sandy Hook, gave only negative results and were of little practical value.

This seems very strange, more especially when it is considered that the fundamental causes are very limited in number. They are:

a. Excessive pressure.
b. Low water.
c. Defective material.
d. Defective workmanship.
e. Local defects.

a. Excessive Pressure.

Excessive pressure can be produced by wilfulness or by careless operation; it can be controlled by autographic recording gauges, alarm whistles, and competent management.

b. Low Water.

Low water may be the result not only of carelessness but of many accidental causes, such as derangement of pumps or injectors, stoppage of pipes, gauge glasses, valves, etc., or even by suddenly augmented leakage, which cannot always be discovered promptly, or even immediately provided for after discovery.

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

c. Defective Material.

Defective material can be readily avoided, and should never be used in any case; it is criminal to use it.

d. Defective Workmanship.

Defective workmanship exists more or less in all boilers, except those which are built without caulked seams and rivets, and punched holes, and all parts of which are prepared and finished by machine-tools and consist mainly of tubes with screwed ends. Defective workmanship develops greater or less defects during service and necessitates constant repair and patching, and may be cause of material weakness in course of time.

e. Local Defects.

Local defects may exist in all boilers initially or develop after a while in service.

a. Determination of Excessive Pressure.

Assuming that a boiler has been properly designed there will be no excessive pressure in service except as previously stated. If all the possible safeguards have not been employed and a record of pressures has not been made, it is still possible to obtain such record for the instant of explosion, by making several tests of strips especially when cut from material of different thicknesses taken from the boiler.

It is a well-known fact that all material used in construction is subjected to stress about one-third of its yield point, but never beyond one-half of this amount. Stress sufficient to cause rupture is always about five times the working stress. If then any part of the boiler had been worn thin by corrosion such material would be overstressed and overstrained.

It is also a well-known fact that excess of stress beyond the yield point invariably raises it above its original value, and if, therefore, careful tests be made of the material, especially with an autographic recorder,* the results will invariably show augmentation of yield point. Different thicknesses will show different augmentations which must always be in direct proportion to

* See the Henning "Pocket Recorder," *Transactions* A. S. M. E., 1897.

the stress applied. If calculations show that the augmentations of yield point in thick and thin material were produced by the same internal pressure, then it is proof positive that excessive pressure existed at the instant of explosion, and also its exact amount. Tests of staybolts will show the same thing. Elongation of the material is, however, reduced by excess of stress, and a second tension test of a piece of material previously ruptured will be very materially decreased. Stress beyond the yield point also changes the shape of the elastic curve in a very characteristic manner.

b. Low Water.

Low water, however caused, always produces excessive heating; and if the temperature rises sufficiently to weaken the material, failure may occur by stripping of the staybolts or rupture of the sheets by bulging between them, or otherwise. If the temperature has raised the material to a low or bright red color, this can be readily determined by superficial inspection. While the fire side will show red rust or a black color, the water or steam side will invariably show a typical steel-blue scale, which will not disappear even after years, as it is a so-called rustless coating. If this be once oiled it will always be distinguishable, even if the plates had been exposed to moisture and gases for years. The color of this scale will depend somewhat upon the temperature at which it was produced, being brightest at those points where temperature was the highest. Carefully made tests, with autographic diagrams, of such material will again demonstrate changes of properties, which are very characteristic. The yield point will be found very low, while the diagram will show a material drop of curve just after the yield point. The elongation will, however, as a rule, be materially increased, with a diminution of tenacity. Nicked and quenched bending tests will again show marked differences between strips cut from the sheet at points which in one case were overheated or were above the low-water line, and in the other were taken from a part below this line. The fracture will also be materially different. To demonstrate the temperature at which the plates happened to be at the instant of explosion, it is necessary to cut strips from points of the overheated plate below the water line. These strips polished on the edges are then held in a clear fire, so that one end remains cold while the

other is heated to a dull yellow or a very bright red. This temperature being reached the bars are withdrawn, and while one is rapidly plunged with one end into a pot of boiling water, the other is allowed to cool in air, but not in contact with wet material or metal or stone. When the piece which has been immersed in boiling water about one inch deep has become nearly cold, below blue heat, it is plunged into cold water.

On the polished edges of both bars will be found scale and heat colors, the temperatures producing them being well established. These bars are then carefully nicked at points opposite every change of color and then broken off at these nicks. By comparing these fractures and their scale and colors with those obtained from pieces cut from the overheated plates, the temperature at which they were at the instant of explosion can be determined with great accuracy. Having thus determined the temperature at which the sheets were during operation, it is also known whether the metal was sufficiently soft to bulge or to strip from the staybolts; examination of plates and bolts will verify the conclusion.

c. Determination of Defective Material.

As it is assumed that all material used in the boiler originally was of good quality, it is only necessary to discuss the changes which might have taken place while in service. It is also assumed that all parts of the boiler are open to inspection at all times, and that, therefore, any local corrosion will be discovered at once and carefully watched during its progress. Ocular inspection is always sufficient for this purpose, and such defects are always visible and easily recognizable before and after failure, and the latter can only occur under the most careless supervision. In service, material may, however, deteriorate by the action of repetitive stress,* the effect of which is again clearly indicated by the results of tests and, more particularly, by carefully drawn diagrams. The yield point will again be changed as before and the elongation similarly. However, different parts of the boiler would show different effects according to their location, and the different action which each is subjected to would produce different results of tests. In the case of excessive stress the differences would be the same in all

* See Martens's "Testing Materials for the Constructor."

material in the boiler. Moreover, a careful study of the results will show other differences which are most fully described in the reference given. These changes of material are, however, produced only by such an immense number of repetitions of stress that they are rarely produced in boilers, which generally wear out and are replaced by new ones before they become distinctly developed.

Long-continued service of material in contact with fire and gases of combustion also produces very marked changes in structure and properties of the materials. The structure becomes more crystalline and brittle. It loses much of its ductility. While the yield point and tensile strength are not much changed, the former generally being lowered, the elongation is decreased materially. Bending tests on both cold and quenched material show considerable deterioration. Nicked bending tests especially produce striking differences. As different parts of the material are affected differently because of difference in temperatures of the fire, they will show different qualities in material and in proportion to these temperatures. This being the case, the cause of such deterioration can be easily traced by careful tests; it is, however, essential that complete records of tests be obtained, which is of course possible only by autographic records. It does not suffice to make the usual determinations of ordinary tests.

It has never been demonstrated that boiler plate, other than that subjected to the action of fire and hot gases, has had its properties changed by long-continued service.

d. and e. Defective Workmanship and Local Defects.

As these must be of very grave importance when sufficiently serious to produce boiler explosions, and are readily noticeable to any trained eye, it is not necessary to discuss them at length; they have often been studied and described. The literature on these subjects is so ample that their enumeration may be omitted.

From the foregoing explanations, it will be seen that careful testing with proper means and apparatus, added to thorough superficial inspection, will always be sufficient to demonstrate the cause of any boiler explosion. It is the failure to apply correct knowledge in individual cases that has led to the frequent conclusion of "cause unknown" or "no reason for accident."

The failure of eminent engineers to reach definite results is due entirely to inadequate investigations, which did not discover the facts available. Moreover, the tests of the material after explosion were made in such a superficial manner, and with such inadequate methods, apparatus, and care, that they were generally valueless.

In my experience I have repeatedly found the results of tests of so striking a nature that this, taken in connection with the knowledge of original properties of the material, gave convincing proof of the causes of explosions. This is not only true of sheets, but also of flues, both of which I have had repeated occasions to study carefully, and there does not seem to exist any doubt whatever that the cause of every boiler explosion can be readily determined, with simple means readily obtainable, by one who has correct and sufficient knowledge of properties of materials as affected by different conditions.

DISCUSSION.

Prof. W. S. Aldrich.—The paper of Mr. Henning is exceedingly to the point. I believe we have all seen the value of practical examples of these changes. It appears to me that he has not gathered together, unless it is under the title of "Local Defects," those which are likely to arise from scale. In marine practice, as well as in stationary boilers in various parts of the country, the question of scale is probably the most vital which confronts the steam-maker. The scale itself and other secondary causes are capable of producing some of the lines of incipient weakness which are referred to in the paper.

The other point of which I wish to speak is in relation to certain work which we found it necessary to do on the *U. S. S. Vulcan* last summer, at Guantanamo Bay, Cuba, the line of work consisting of jacking up the furnaces of the *U. S. S. Indiana* and the *U. S. S. Detroit*. Some of the furnaces of these battleships, owing to the use of salt water under the exigencies of blockade duty off Santiago de Cuba, had come to be down in the flats or cylindrical portion, as well as in the corrugations. The *Vulcan's* work in their behalf consisted in jacking the furnaces back to the original position. The crowns were first heated to a cherry red and then jacked back by means of hydraulic jacks, with the use of cast-iron former plates made on board the *Vulcan*. The point

to which I wish to call the attention of the writer is this: will not excessive jacking of these marine furnaces in time result in serious deterioration of the material, quite apart from that due to the fire and the hot gases? since Mr. Henning states "it has never been demonstrated that boiler plate, other than that subjected to the action of fire and hot gases, has had its properties changed by long-continued service." Such jacking of boiler furnaces is an exposure to an artificial strain entirely, and quite different from that naturally arising in everyday service.

It is the case that a large number of the coastwise steamers jack their boilers sometimes every trip. Their furnaces to a greater or less extent are said to be not injured by that process, and I should like to inquire if there is any experience on this question, because it appears to be a vital one. Very fortunately the specifications for the mild steel of those boilers in the naval vessels called for the very best material, and the furnaces of the *Indiana* and the *Detroit* stood the jacking process without any visible marks of deterioration.

Prof. Forrest R. Jones.—About a year and a half ago a boiler of a locomotive in passenger service on one of our Western roads exploded, with somewhat disastrous results to both life and property. An examination of the plates after the explosion showed that they were badly grooved and pitted, and that the grooves, some of which were of peculiar form, being very narrow and deep, ran more than two-thirds of the way through the plate, and seemed to be ample cause for the rupture. The locomotive was standing in the round-house, almost ready to take out a passenger train, the steam pressure being up, so far as was ever learned, to about the normal; about 4 or 5 feet in width of the cylindrical portion of the shell was torn out and thrown some 500 feet. The grooves and pitting were shown very distinctly in this portion, some of the grooves running circumferentially and others longitudinally with the boiler. The matter was brought into court some two weeks ago, and there were developed some points as to the inspection and care of boilers, and it was shown that this particular company did not inspect the boiler shell inside throughout except when the flues had to be taken out for repairs, and that the flues of this particular boiler had not been removed so as to allow inspection inside throughout for some four or five years. The same method of inspecting, or not inspecting, was shown for some of the other principal railway lines running out of Chicago.

A piece of the plate, a rather small piece, was tested in order to learn its qualities. The boiler, I might say, had been made in 1867, and was of wrought-iron plate. A large portion of the plate that had been originally put in the boiler was still there. The longest piece that could be gotten from the fragmentary plate was 12 inches, and therefore the elongation and corresponding properties had to be measured in 6 inches of length. The material showed an elastic tensile strength of 24,000 pounds to the square inch, about 44,600 pounds maximum tensile strength, and the total elongation in 6 inches was less than 2½ per cent. It was about ⅝ of a square inch in sectional area. It would bend to an angle somewhat less than 60 degrees before cracking, and would not bend as far as 90 degrees before breaking completely in two. The direction, relatively to the fibre, in which the test for tensile strength was made was difficult to determine. The plate, being made of wrought iron, was laminated, of course, being made up of various plates welded together. There was one layer on each side that appeared to be of good material, which comprised about one-fourth of the thickness of the plate. The remainder of the plate seemed to be made up of several laminæ which were welded together with more or less dirt, scale or oxide between them. An endeavor to find the direction of the grain was unsuccessful. This was done by etching, but it appeared, as nearly as could be told, that the outside plate or layer on one side had the grade running at right angles to that on the opposite, so that the plate might have been as strong in one direction as another. The smallness of the piece would not allow tests in both directions, however.

The great length of time which was allowed to elapse without inspecting the interior of such an old boiler—these grooves which I mentioned were on the interior—leads one to think that if possible there should be some method of getting inspections more often upon locomotives which are to draw passenger trains. Of course it would be a very difficult thing to do that. As to this grooving and the length of time it would take to occur, there were various opinions expressed as testimony in court, as to the length of time the grooves could have been seen before the bursting of the boiler. An inspector for the Hartford Steam Boiler Inspection and Insurance Company, who had inspected for twelve years not less than a hundred boilers a month, said that the grooves would have been visible to an extent sufficient to have caused a careful person to

watch it at least ten years before. Another testified that it would have been visible to an inspector on the inside at least five years before. The groove was of a very peculiar form, at least to me. The plate had apparently been eaten down or corroded about one-third of the way through. This corrosion extended back some distance from the seam. Then, close along the seam, the groove was a very sharp one, appearing as if it had been formed almost as a crack, gradually extending downward into the plate, but corroding as it extended downward. The result was that in the exploded boiler the plates showed this groove to be narrow and deep, extending about one-third of the way through the plate, about $\frac{1}{8}$ of an inch wide at the top, running down to a sharp groove at the bottom, all corroded, so that in order to tell the depth of the groove when the plate was not fractured it was necessary to put a sharp point in, as of a pen-knife. Such measurements showed that the thickness of the remaining material was less than one-third of the original thickness of the plate; that is, less than $\frac{1}{8}$ of an inch.

I should be very much interested in hearing how wrought-iron plates, such as were used for boilers, were made as long ago as thirty-two or thirty-three years, for those plates must have been made that long ago to have been in this boiler.

Mr. Henning.—Professor Aldrich's surmise that I have included "the effects of scale" in "local defects" is quite correct. They are visible to any trained eye. I should like to refer to the effect of scale aside from pitting, grooving, etc., and point out the difference in the appearance of the plates within the boiler in the presence or absence of scale.

If the crown sheet, for instance, becomes red hot and the steam comes in contact with it, a distinctly characteristic blue scale will be formed, which is shown in some of these test pieces, and that blue scale is never found when there is scale on the metal. Whenever there is scale on the metal there will be a red rust, and generally a velvety brown rust which either sticks to the scale or to the steel when the scale is removed; but the particular blue scale that is formed with low water when the metal is red hot and the steam comes in contact with it is quite characteristic. I do not think it can be formed, because it requires steam in contact with red-hot iron to produce that non-corrosive scale. That scale is so non-corrosive—it is really the same thing as the non-corrosive scale produced by the Bower-Barff process—that it will remain for

years on a boiler that has been standing in the round-house where engines have been blown off and fired up, and I have known cases where that scale has been perfectly clear and self-evident after five years and so plain that it just looked as though a painter had painted one part of the sheet between two sets of staybolt holes, on opposite sides, the most perfect blue, the so-called steel blue, while either side of that band the mill scale was still on the plate, but rusty; but as soon as the rust was cleaned off the original mill scale became visible. It is so characteristic that once you know it you cannot fail to identify it. That scale does not appear under the boiler scale, such as deposits of lime and other substances. It can only appear when the steam comes in contact with the red-hot metal. In reference to jacking up furnaces after the crown sheets have settled, I wish to say that it is an expedient which will do after a battle, and until you can get to a navy yard and pull out that crown sheet and put in another one. But such crown sheet is injured by such treatment, and why is easily explained. Let us take a stress-strain diagram of steel—I will not speak of iron—and it is always characteristic that the elastic line OY_1 is nearly straight up to a certain point, Y_1, and then above that point the line begins to fall off very rapidly in the horizontal direction because the elongation increases rapidly (Fig. 224). Now, if steel be tested as it comes from the rolls and has not been treated, its curve at the point called the yield point will be very similar to that represented by OY_1. The part OY_1 is the elastic curve. If the steel had been red hot this same line would have taken a shape more nearly like that shown by OY_2; that is, it would have dropped down a little, probably, and then gone up very gradually, while in the first case that corner at R would have been rather sharp. If that material in the crown sheet had been jacked up cold, we should find a new yield point, which is at the point Y_3. Now, apparently the material has a higher elasticity than it had at first, but the actual strength of the material will be still about the same as it was before, but it will not elongate as much as it did before, and if it is jacked up again we will again raise the yield point up to that point, Y_3, and the curve beyond Y_3 becomes more nearly straight, and we will again have nearly the same strength of material, but the curve will be shortened at 3, and if we do that jacking up frequently the steel will be ruined, and likely to give trouble. If the furnaces were first heated and then jacked up hot the sheets would in each case be stretched, and their ductility lessened. As

the heating would be only local (in spots) excessive stress would be brought on the sheet where the color is no longer red, but blue, and it is a well-known fact that it is dangerous to work steel at that heat. The stress-strain diagram of the red-hot material would resemble $OY_2 2$, showing weaker and softer material, while the diagram of the blue-hot steel would resemble $OY_3 3$. In either case the steel would suffer injury, and it is only a question of time, or number of repetitions of such treatment, to injure the furnaces irreparably thereby.

As the corrugations in the furnaces mentioned are made while

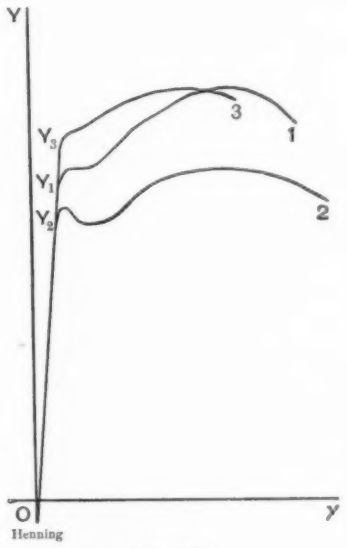


FIG. 224.

the steel is at a red heat, the stress-strain diagram of the steel would be like $OY_2 2$; the steel would be softer, have a lower yield point and resistance, but greater elongation than in its original condition, and it is not in quite as good a condition as plate in the condition in which it comes from the rolls. Iron which has been in use a long time, subject to heat, will behave as shown by $OY_3 3$, except that its tensile resistance will be decreased. That is, if it be tested after a long period of use it will fail, showing very slight elongation. Iron was, of course, made in different ways; some was rolled out of properly rerolled iron carefully piled, while some was simply rerolled cover plates, then filled up with scrap, producing

a good surface, of good ductile material, so called, which is ductile but which would not wear well in service. Old boilers, very many of them, at least, were made of such material; it is very rare to find any iron that was made thirty years ago in service in any boilers. I have never seen anything like it. But all of these effects can be distinctly traced by a careful test. They have been so well studied and so clearly defined that all that is necessary to do is to make a sufficient number of tests of different parts of the same boiler, and the evidence adduced from such tests will generally, in fact I think almost invariably, explain what has happened to the material.

Prof. Forrest R. Jones.—The material which was spoken of as coming from the exploded boiler, and which was tested by tension and bending, had not been subjected to the heat of the furnace, since it was part of the cylindrical portion of the boiler ahead of the wagon top and in front of the fire-box, coming from the side of the boiler; that is, not just at the bottom. It came from the water portion of the boiler. It was not in contact with the fire at all.

Mr. W. M. McFarland.—I think it would be well for Mr. Henning, on account of one statement which he has made, to define it a little more clearly. I understood him to say, in reply to Professor Aldrich's question, that the plan of jacking up a furnace by heating the part that had bagged and setting it out would do as a temporary expedient until you could get to a navy yard and take the furnace out, from which I infer that his idea is that if the furnace had bagged once that was the end of it. So far as my experience goes that is by no means the case, and from his own remarks a little later, where he spoke of successful jackings of a furnace, evidently he did not mean it either. Inasmuch as he has investigated the subject so thoroughly, I think it would be well if he would give some idea as to how many times he thinks the operation could be performed with reasonable assurance of safety.

Mr. Henning.—I did not intend to intimate that one jacking up or reheating to a very small amount had that effect. I said that repeated reheating and repeated jacking up would gradually produce this condition. The amount of jacking up, or the number of times that the space between staybolts can be jacked up, depends on the amount of deflection that has to be rectified. It is not possible to make a general statement in any case as to how often it can be done. It depends on many conditions. Every

time the furnace is jacked up the metal is injured more or less, and seams and rivets may also be loosened. How often that can be done properly I cannot say. I did not intend to say that you would have to take the furnace flue out if it was done once, but only if it is frequently done with the crown sheets by simply putting a jack under the low part and jacking it up.

Mr. McFarland.—Perhaps we are thinking of different things. You were evidently speaking of a locomotive fire-box. I had in mind a marine corrugated furnace; that was what I was talking about. So far as I know with regard to the corrugated furnaces in marine boilers, this process of jacking up is quite a frequent thing.

Mr. Henning.—My statement refers to any material which is treated in that manner. If material is strained in such a manner as to change its shape permanently its quality will always be injured. If a furnace, whether corrugated or staybolted, has changed its shape the material can never again be upset so as to bring it to its original condition. The section of some parts of the corrugations under treatment and some parts of the boiler will also be strained, and this will produce injurious effects in any material, which cannot be done often without serious injury. If the sheets are heated locally the effect is equally bad. If an entire flue were heated simultaneously the injurious effects would be slight, under moderate temperatures; but it is always a thing which tends to injure the boiler. It makes no difference what kind of a boiler it is. I am speaking of the effect on the steel, while the effect on the boiler is something that can be demonstrated by superficial inspection, which is a totally different thing. Although the treatment described affects the steel injuriously, the boiler as a machine is neither destroyed nor even made immediately dangerous. Frequent similar treatment will no doubt produce noticeable effects, which would become more pronounced were it not for the fact that an ample factor of safety is provided in the original design of the furnace. It is possible that the original method of manufacturing corrugated furnaces sometimes materially weakens the steel. But proper testing would demonstrate such effects. It may be necessary to jack up low spots in corrugated furnaces, but it certainly cannot do otherwise than gradually injure the material.

It is my opinion that the furnaces of the *Indiana* and *Detroit* "had come to be down," because salt deposited on the low parts of corrugations and flat portions allowed the furnaces to become

red hot in spots. If for any reason a large area had become red hot there would have been failure of the furnaces. Those parts of the furnaces which did not become red hot supported the weakened spots, and their surplus strength, due to large factor of safety, prevented actual collapse. It is possible that a small amount of local bulging of the furnaces causes the scale or salt deposit to crack off, or water to flow in and thus cool off the metal before actual failure had taken place. This action is possible. The overheating of small areas of course injures the steel if it reaches a high degree. The subsequent jacking after heating adds additional injury by local reheating and by forcing the material back into its approximate position. Such necessary deformation cannot be anything but injurious, not only because of necessary deformation of the furnace, but also because of actual injury to the metal. Reheating steel without subsequent uniform reworking always weakens boiler plate, and it is only a question of frequency of such treatment before palpable injury will arise. I have no hesitation in saying that most positive measures should be taken to prevent such downward bulging of corrugated furnaces. It is of course true that a great many tricks are employed in practical management of steam boilers, which, although injurious to their life, are almost necessary to keep the ships in running order and earning money. On the other hand, if boilers are designed and operated in such manner that such treatment becomes unnecessary, there can be no question that they would remain in good service for an indefinite period.

The short life of marine boilers is notorious. Rough treatment causes deterioration and necessitates constant watchfulness and attention. It will hardly be denied that land or inland boilers remain in service for a much longer period than marine boilers, and no doubt the hard usage the latter get is largely responsible therefor.

DCCCXV.*

THE POWER PLANT OF A UNIVERSITY.

BY EDWARD A. DARLING, NEW YORK CITY.

(Member of the Society.)

INTRODUCTORY.

ONE of the foremost industries in the United States to-day, as well as the most widely spread, is that which is commonly known as the "higher education."

From the earliest days of the settlement of this country this industry has been most carefully fostered and encouraged, and its growth has been normal at all times to the growth of the country.

It is no longer a thing apart, cloistered in monastic seclusion, wrapped up, for the most part, in questions of dogma or abstract philosophy, and emitting from time to time flashes of light of more or less illuminative value to the mass of mankind, but it is to-day, in this country, a corporate part of the body politic, instinct with the life of the people, alert to respond to the needs and requirements of a rapidly moving age, and throbbing with a patriotism that rarely fails to respond in time of trouble.

It has received the countenance and support of men of every shade of religious and political opinion. It has been nurtured and enriched by private munificence and subsidized by national and State legislation, and is to-day accounted one of the chief bulwarks of the state, as well as the foremost of the forces that make for the upbuilding of the commonwealth.

That gifts to universities and colleges have not always proved unmixed blessings has been largely due to two causes. The first was the common mistake of trying to get "too much for your money," as evidenced by the erection of buildings of undue dimension, and of more or less flimsy construction, which become, in time, burdens on the institution owning them, through the moral obligation imposed of keeping them in use and repair.

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

The second was that endowments for professorships and lectureships were invested in funds or real estate which bore too high a rate of interest to be stable, or the gifts were so specific in character that they could not be applied to other needs, when the original objects were no longer considered worthy of attainment.

The testimony of a large majority of the boards of management of institutions of learning to-day would be that the most useful gift is that which is least specific in character.

It has been a universally recognized principle in the administration of the higher education that the institution offering it should be as independent as possible of the fees of its students, to the end that the quality of the instruction should not be influenced by the need of increasing the number of fees received.

The same law that in the commercial world influences people to buy where they can get the most and the best for their money is at work in the educational world, and where the intending student is not deterred by sentimental or family considerations, he, or his guardian, selects, as his future *alma mater* that institution which makes the best tender to him, counting carefully the quantity and quality of its educational advantages, the prestige of its degree, the cost of tuition, and of living while in attendance, the distance from his home; and last, but not least, it must be said, its athletic and social possibilities, so important in the mind of the intending student.

The universities of America are trusts, existing for the avowed purpose of giving to the student as much education as possible—more than he has paid for.

The same causes which make for success or failure in the commercial world operate here, and should be as carefully considered as where the end in view is the purely selfish one of earning dividends.

It may be that these educational trusts will not attain their full measure of usefulness until they come to realize the meaning of an industrial trust, with its combination of capital, its wise conservation of energy, and its avoidance of unnecessary and ruinous competition.

Of course it would not be expedient, nor would it be necessary to carry the process of combination to the extreme limit which it has reached in some of the industries, but surely there are opportunities for consolidation in the case of groups of universi-

ties, colleges and professional schools, almost within sight of each other, working at cross purposes to similar ends, wasting to a certain extent their own energies and the funds of those who support them, doing this unwittingly and unselfishly too, for it often means hardship to those engaged in such fruitless competition.

This process of combination would not only inure to the benefit of those most directly concerned—the faculties and students—but would tend to raise the standards of scholarship and of professional training throughout the whole country.

A recent example of consolidation along somewhat similar lines might be cited in the uniting of the Astor, Lenox, and Tilden foundations into the New York Public Library.

Competition may be the life of trade, but it can easily be the death of progress in education. Its application, excepting at long range, should surely be as unnecessary in the propagation of the higher education as it would be in carrying on the postal system or in providing water supply for a community.

The varied activities of a modern university make necessary the operation and maintenance of a library, a comprehensive system of lecture and recitation halls, laboratories, and workshops, which are in themselves so many industrial enterprises on a small scale; a system of museums, hospitals and clinics; an astronomical observatory; a printing and publishing establishment; an experimental farm, arboretum and botanical garden, equipment for scientific expeditions and field work; a chapel and academic theatre; a gymnasium, with its attendant features, baths, athletic field and boathouse; a refectory, dormitories and other buildings devoted to the informal and social life of its students, and in addition to all these the necessary plant to furnish light, heat, power, ventilation, water, gas, etc., wherever needed.

The personnel of a university includes besides its president, board of trustees or governors, and the officers of instruction, who form the "line," a body of administrative officers which may be called the "staff," and a force of petty officers and enlisted men, which includes clerks, engineers, mechanics, janitors, porters, messengers, firemen, gardeners, laborers, etc.

The government of an enterprise of this kind must be carried on with an eye single to the main purposes in view—to husband its resources and increase its capital and plant to meet the in-

creasing demands of latter-day civilization; to recruit carefully from the best material obtainable its teaching and administrative force, men and women who will in turn seek to add strength to the institution in every way possible.

To foster and encourage instruction and research in branches which do not appeal to popular favor, but which, none the less, make for the well-being of the body politic.

To stimulate and help those who hunger for education, but lack the means of paying for it, without encouraging the evasion of just payment by those who can and should make due return for their education.

To maintain an impartial attitude toward all the professions to which it sends recruits, where there is the constant tendency on the part of one or another of these professions to secure, through its representative in the teaching force, an undue amount of attention and an unfair share of its funds, at the same time giving to each professional school the greatest possible latitude for development and usefulness in its own peculiar field. In short, to increase the scope of the institution, and add to its usefulness in every way possible.

The economic relation which exists between the teaching force of a university and its material plant is very much the same as that which exists in an industrial establishment between the working force it employs and the equipment which it operates.

A manufacturing concern which would hold its own in this age of keen competition must not only employ the very best engineering and administrative talent, and the most intelligent and efficient labor, but it must also have well-arranged, commodious buildings, and machinery which embodies the very best and latest practice in its particular field.

To neglect either the *personnel* or the *matériel* is to court gradual decay and ultimate defeat.

In all relations between mind and matter a nice balance is difficult to maintain, but it should none the less be striven for, and to ignore or underrate the full value of either part of the equation is the part of unwisdom.

The operation of a university on the material side is becoming more and more a matter of concern to those who direct its destinies and who have to provide the funds. To keep down the fixed charges in what may be called its housekeeping department to the closest limit consistent with efficient service; to

maintain its equipment at the highest state of efficiency; and to secure the erection of buildings best suited to the purpose for which they exist, and which shall, by their symmetry and dignity, give architectural expression to this purpose—these are all questions of the very first importance.

In planning and operating its buildings and equipment, advantage should be taken, not only of the experience of other universities, but of the experience of those who may have worked out solutions of some of their problems, with perhaps greater success, in the commercial and industrial world.

From recently compiled statistics it appears that the colleges and universities of the United States carry on their rolls 165,000 students, and employ 10,000 teachers.

They have an aggregate capital of three hundred millions of dollars, and their annual disbursements amount to fifty-two millions.

Great as these figures seem, they are relatively small when we consider the vital importance of this interest to the whole country, and when we remember that a single railroad corporation has an invested capital more than twice as large as all the universities and colleges together, and when we consider the yearly budget of the city of New York alone to be ninety millions of dollars.

It is the purpose of this paper to describe the design and installation of the engineering plant of one of our universities, to show in outline the building equipment it serves, and to give a few pages from the log-book of the operation of this plant.

If the telling shall prove of interest to the members of the Society, and be of some small profit to those directly interested in similar enterprises, its object will have been attained.

COLUMBIA UNIVERSITY.

The history of this university is not unlike that of others among the older universities of this country.

Like these, it began in a small way, with a single professor, who was also its president, instructing a half score of students, without an endowment or even a home of its own. And it has gradually grown as others have grown, until to-day it numbers its teachers by the hundred, its students by the thousand, and measures its capital by the million.

It counts among its graduates men who have left a mighty impress on affairs of state and church and science.

During the period from its foundation as King's College, in 1754, until the breaking out of the Revolution, it showed its loyalty to the Crown by sending out twelve graduates who became officers in the British army, and one who became an admiral in the British navy. These men, in time, did their part toward the expansion of what is now the British empire.

When the Revolution came twelve others clave unto their native land and became officers of the Continental army, five others were delegates to the Continental Congress, and in the stormy period that immediately followed the Revolution three others, Hamilton, Jay and Morris, exercised a paramount influence in the formulation and ratification of the Constitution of the United States.

During this period the name of the institution was changed from King's College to Columbia, the first time that name was made use of in history.

General Washington, who we should be proud to remember was an engineer, lent his aid to the rehabilitation of the college, giving his surveying instruments to form a beginning for an engineering equipment, and entering his stepson as a student.

In the early part of the present century three of its graduates took a hand in the development of engineering in this country. John Stevens introduced the steam railway and the screw propeller, Chancellor Livingston furnished the funds for Fulton's experiments, and De Witt Clinton cut the Erie Canal.

During the years that have intervened between that time and this, Columbia College has paralleled the work of its sister universities and colleges with varying success.

COLUMBIA UNIVERSITY, FIFTY-NINTH STREET SITE.

DEPARTMENT OF BUILDINGS AND GROUNDS.

POWER HOUSE LOG FOR WEEK ENDING APRIL 2, 1899.

		Boilers.							Coal, Gross Tons.	Ashes, Gross Tons.	Feed Water, Cubic Feet.	Average Temperature.		Barometer.	Inches Draft.	Weather.	Engines and Dynamos.				Ampere Hours.	Storage Battery.	Feed Pump No. 1.	Feed Pump No. 2.	House Pump No. 1.	House Pump No. 2.	Filters.	Brine Pump No. 1.	Brine Pump No. 2.	Ice Machine No. 1.	Ice Machine No. 2.	Feed Heater No. 1.	Feed Heater No. 2.	Cylinder Oil, Gallons.	Engine Oil, Gallons.	Lbs. of Waste.	Total Water, Cubic Feet.	Labor, Cost.	Notes.
		1	2	3	4	5	6	7				Outside.	Inside.				1	2	3	4																				
March	27	Monday	....	....	....	B. 7 17	B. 7 17	24	24	9.25	1.25	3,340	36	85	30.7	.45	F.	14	14	....	....	8,484	+4	....	24	24	....	W.	....	24	....	7	24	....	1	1.25	2	10,450	$17.10	
"	28	Tuesday	....	....	....	B. 7 17	B. 7 17	24	24	8.5	1.05	3,300	38	85	29.6	.42	F.	14	15	....	....	8,700	....	....	24	24	....	W.	....	24	....	7	24	....	1	1.2	2	9,264	17.10	Replaced broken arm on brine pump No. 2.
"	29	Wednesday	....	....	....	B. 7 17	B. 7 17	24	24	9.6	1.33	3,484	37	85	29.3	.43	S.	14	15	....	....	8,671	....	....	24	24	....	W.	....	24	....	5	24	....	1	1.25	2	8,120	17.10	
"	30	Thursday	....	....	....	B. 7 17	B. 7 17	24	24	8.5	1.25	3,048	40	90	29.6	.44	C.	14	15	....	....	8,633	....	....	24	24	....	W.	....	24	....	8	24	....	1	1.4	2	13,170	17.10	
"	31	Friday	....	....	....	B.24	B. 7 17	24	24	6.4	.87	2,484	46	96	29.55	.41	F.	14	15	....	....	6,148	....	....	24	24	....	W.	....	24	....	7	24	....	1	1.5	2	12,060	17.10	Holiday.
April	1	Saturday	....	....	....	B.10 14	B.10 14	24	24	9.25	1.25	3,300	35	82	29.7	.45	F.	14	15	....	....	7,428	....	....	24	24	....	W.	....	24	....	6	24	....	1	1.25	2	9,405	17.10	
"	2	Sunday	....	....	....	B.24	B.10 14	24	24	6.9	1.00	2,630	34	84	29.9	.40	R.	15	15	....	....	6,000	....	....	24	24	....	W.	....	24	....	3	24	....	1	1.2	2	8,360	17.10	
		Summary and Averages	....	....	....	B.86 82	B.55 113	168	168	58.4	8.00	21,486	38	87	.....	....	....	99	104	....	....	53,997	....	...	168	168	....	....	....	168	....	43	168	....	7	9	14	70,829	119.70	

REMARKS:

The figures given under "Boilers," "Engines," "Pumps," etc., indicate hours run.
B. under "Boilers," means banked.
W. " "Filters," " washed.
F. " "Weather," " clear.
C. " " " cloudy.
S. " " " stormy.
R. " " " rain.

COLUMBIA UNIVERSITY, 116TH STREET SITE.

DEPARTMENT OF BUILDINGS AND GROUNDS.

POWER HOUSE LOG FOR WEEK ENDING APRIL 2, 1899.

		Boilers.										Coal, Tons.	Water, Gallons.	Ashes, Tons.	Average Temperature.		Barometer, Inches.	Draft, Inches.	Weather.	Engines and Dynamos.						Ampere Hours.			Feed Pump No. 1.	Feed Pump No. 2.	House Pump No. 1.	House Pump No. 2.	Elevator Pump No. 1.	Elevator Pump No. 2.	Vacuum Pump.	Fire Pump.	Air Compressor.	Filters.	Feed Heater.	Forced Draft.	Cylinder Oil, Gallons.	Engine Oil, Gallons.	Lbs. of Waste.	Water, Cu. Ft.	Labor.	Notes.
		1-2	3-4	5-6	7	8-9	10-11	12	13-14	15-16	17				Outside.	Inside.				1	2	3	4	5	6	Power.	Light.	Total.																	
March	27	Monday			9 / 12	24	24 / 24						29.95	73,717	6.2	47.2 / 23.8	78	30.7	.75	C.		5.45		8	4.30		9,490	5,755	15,245	24		16.5		11.5		8.5		10		On		1	1.25	1.75	14,520	$33.76	
"	28	Tuesday			11 / 11	24	24 / 24						26.38	65,118	6.65	41.9 / 28.2	77	29.6	.60	R.	2.45	7.40		6			6,870	9,100	15,970	24		16.5		11.5		8.5		10	W.	"		1	1	1.75	13,700	33.76	
"	29	Wednesday			11 / 11	24	24 / 24						25.05	61,600	3.5	43.7 / 30.0	78	29.3	.66	F.	2.40	7.40		6			7,410	8,250	15,660	24		16.5		11.5		8.5		10		"		1	1	1.50	8,300	33.76	Swimming pool filled—93 hours.
"	30	Thursday			5 / 11	16	24 / 24						22.33	65,017	3.5	47.5 / 31.8	80	29.6	.69	F.	2.45	8.40		5			7,075	8,130	15,305	24		16.5		11.5		8.5		10	W.	"		1	1	2	5,300	33.76	
"	31	Friday					12 / 24						10.65	25,700	1.95	54.8 / 37.2	80	29.55	.70	F.																								2,300	33.76	Holiday. Received coal.
April	1	Saturday			11.5 / 11.5	15	24 / 24						21.43	52,700		50.6 / 31.1	81	29.7	.72	F.		8.30			7.40		4,030	5,275	9,295	24		16.5						10		"		1	1	2	4,900	32.10	Centred No. 2 and No. 4 Engines to trammel.
"	2	Sunday					24 / 24						11.66	28,500	4.5	49.0 / 27.0	82	29.9	.65	F.																			W.	"				1,300	32.10	
		Summary and Averages			47.5 / 56.5	103	156 / 168						147.45	372,532	25.3						.05	38 / .15		25	12 / .10			71,375	168		82.5		46		34		50				5	5.25	9	50,120	233.00	

REMARKS:

The figures given under "Boilers," "Engines," Pumps," etc., indicate hours run.
B. under "Boilers," means banked.
W. under "Filters," means washed.
F. under "Weather," means clear, "C" cloudy "S" stormy, "R" rain.

Although it has been called Columbia College until very recently, it has almost from the first exercised the functions of a university, in graduating men in the learned professions; and its schools of medicine, law, political science, philosophy and mines, have long been among the foremost in this country.

Like its neighbor, the University of Pennsylvania, it has always carried on its work within the confines of a great city, and has been compelled to move its site from time to time as the city grew and compelled it to seek more roomy quarters.

While these removals would seem to be a disagreeable and wasteful necessity, on second thought they appear to work a great advantage, in that they permit a renewal of youth on the part of the institution, and the erection of buildings more suitable to the needs of succeeding generations.

The university proper is now occupying its third site, and the medical school has had no less than five removals.

In 1887 the medical school, or more properly the College of Physicians and Surgeons, was moved into its present spacious quarters in Fifty-ninth Street, and just ten years later the other departments of the university took possession of the new site on Morningside Heights.

The College of Physicians and Surgeons is now located opposite a great hospital, in a district best suited to its activities, and the university proper is seated on a hill-top, in a region of quiet, where the currents of traffic are not likely to disturb it for many generations to come.

Both sites are on Amsterdam Avenue and from the clock tower of the College of Physicians and Surgeons the dome of the library looms up three miles away to the northward.

The present home of the College of Physicians and Surgeons was almost wholly provided for by gifts from different members of the Vanderbilt family, namely, the purchase of the site and the erection of the original building by William H. Vanderbilt, and the subsequent additions of the clinic, the Institute of Anatomy, and the Sloane Maternity Hospital, which were provided for by the sons, the son-in-law and the daughter of Mr. Vanderbilt.

The present dean of the college, Dr. James W. McLane, had been the means, in the first instance, of enlisting this powerful interest in the upbuilding of the school, and since its reincorporation as an integral part of the university, which took place in the

second year of President Low's administration, it has reached the very front rank of the world's medical schools, with an equipment which is at once the most modern and most complete in existence.

A ground plan and a view of the buildings as they stand to-day, taken from the grounds of the Roosevelt Hospital opposite, are shown in Figs. 225, 226, 227.

The lines of the buildings as they were erected in 1887 are hatched inside the lines of the completed structures, and give some idea of the rapid growth that has taken place, a growth that was hardly foreseen, for it might, perhaps, have been placed to better advantage. The experience gained along this line by the trustees was, however, soon to be made use of in planning for the new university.

The buildings are of a plain and most substantial character, and do credit to the skill and good judgment of the architect who designed them, Mr. W. Wheeler Smith. The total investment for buildings and equipment is something over two and a half millions of dollars.

When the original buildings were erected the power plant, which was located in the basement of the principal building, consisted of a battery of five horizontal tubular boilers, working under 40 pounds steam pressure, which furnished steam for heating and for the operation of a 300-light, belted bipolar generator, seven ventilating fans, a steam laundry, and a single hydraulic elevator.

Th generator was run from 4 until 10 P.M., during the winter months, for lighting the large lecture rooms and dissecting room in the main building.

The only call for steam all the year round was for operating the laundry in the hospital. Simple as this plant was, it was then the most complete of its kind in this country.

When, in 1891, the medical school became a corporate part of the university, and it then began to share the benefits of the general endowment, the consequent extension of the curriculum made necessary the extension of buildings and equipment.

This extension covered a period of six years, and, beginning piecemeal, it was difficult to see at first the lines along which the growth would take place.

The problem which this growth presented, of providing for the physical needs of those who carry on modern medical education,

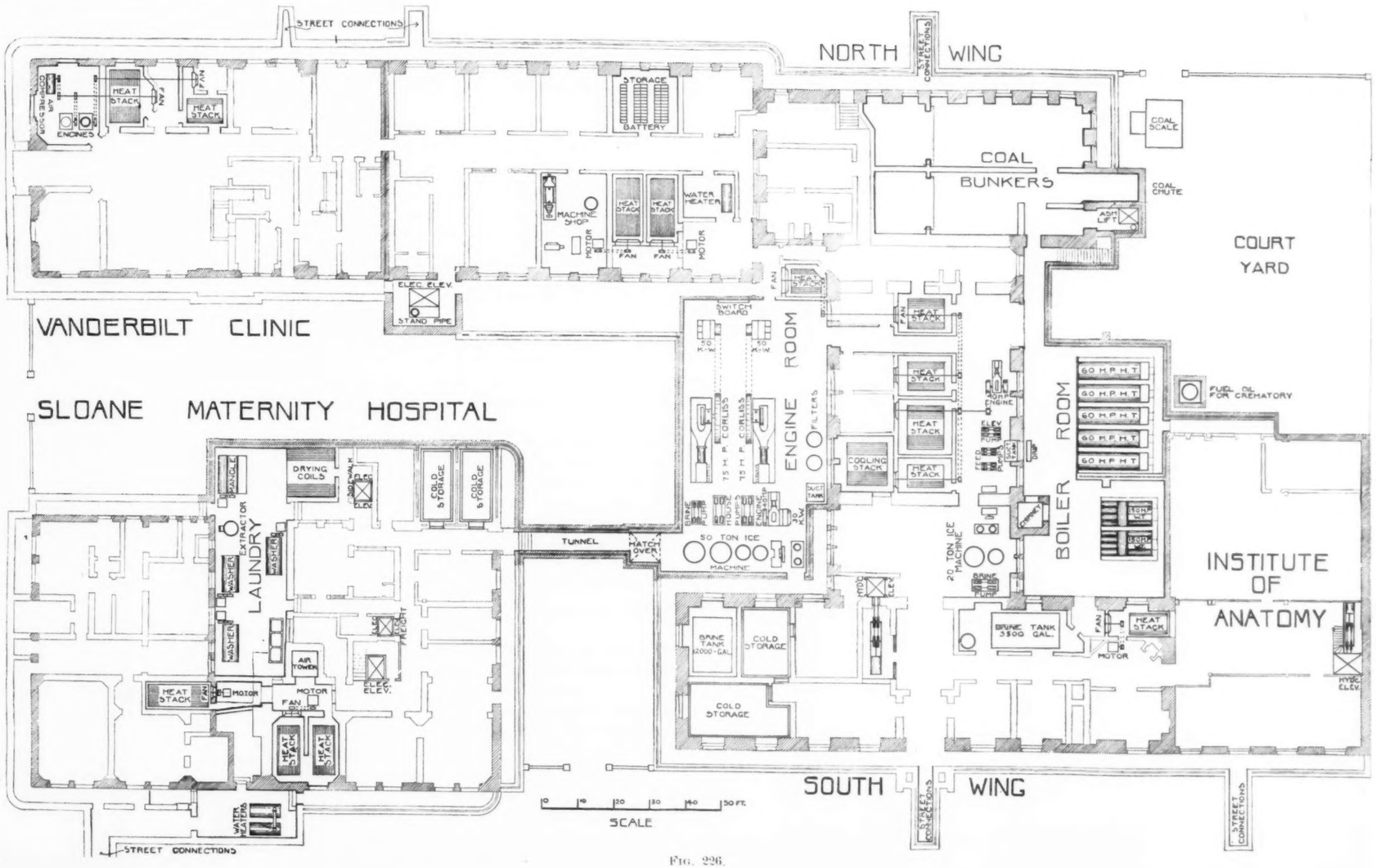
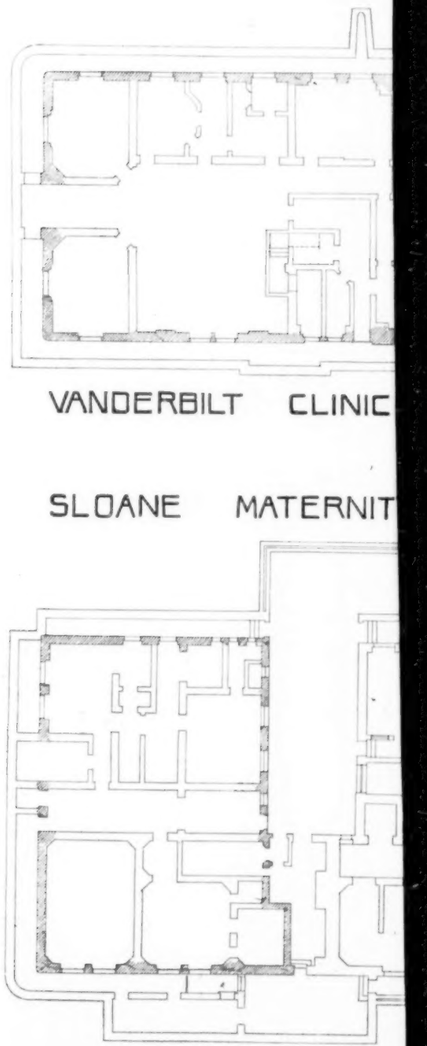

STREET CONNECTIONS
NORTH WING
STREET CONNECTIONS
COAL SCALE
COAL BUNKERS
COAL CHUTE
ASH LIFT
COURT YARD
STORAGE BATTERY
ENGINES
HEAT STACK
FAN
MACHINE SHOP
MOTOR
WATER HEATER
ELEC. ELEV.
STAND PIPE
VANDERBILT CLINIC
SLOANE MATERNITY HOSPITAL
SWITCH BOARD
ENGINE ROOM
75 H.P. CORLISS
FILTERS
COOLING STACK
FEED PUMPS
DUCK TANK
BRINE PUMPS
50 TON ICE MACHINE
20 TON ICE MACHINE
BOILER ROOM
60 H.P. H.T.
10 H.P. L.T.
3 H.P. L.T.
FUEL OIL FOR CREMATORY
INSTITUTE OF ANATOMY
BRINE TANK 3500 GAL.
BRINE TANK 2000 GAL.
COLD STORAGE
HYDR. ELEV.
TUNNEL
MATCH OVER
LAUNDRY
DRYING COILS
EXTRACTOR
WASHER
DRY SINK
COLD STORAGE
AIR TOWER
ELEVATOR
WATER HEATERS
STREET CONNECTIONS
SCALE
0 10 20 30 40 50 FT.
SOUTH WING
STREET CONNECTIONS
STREET CONNECTIONS

FIG. 226.


VANDERBILT CLINIC
SLOANE MATERNIT

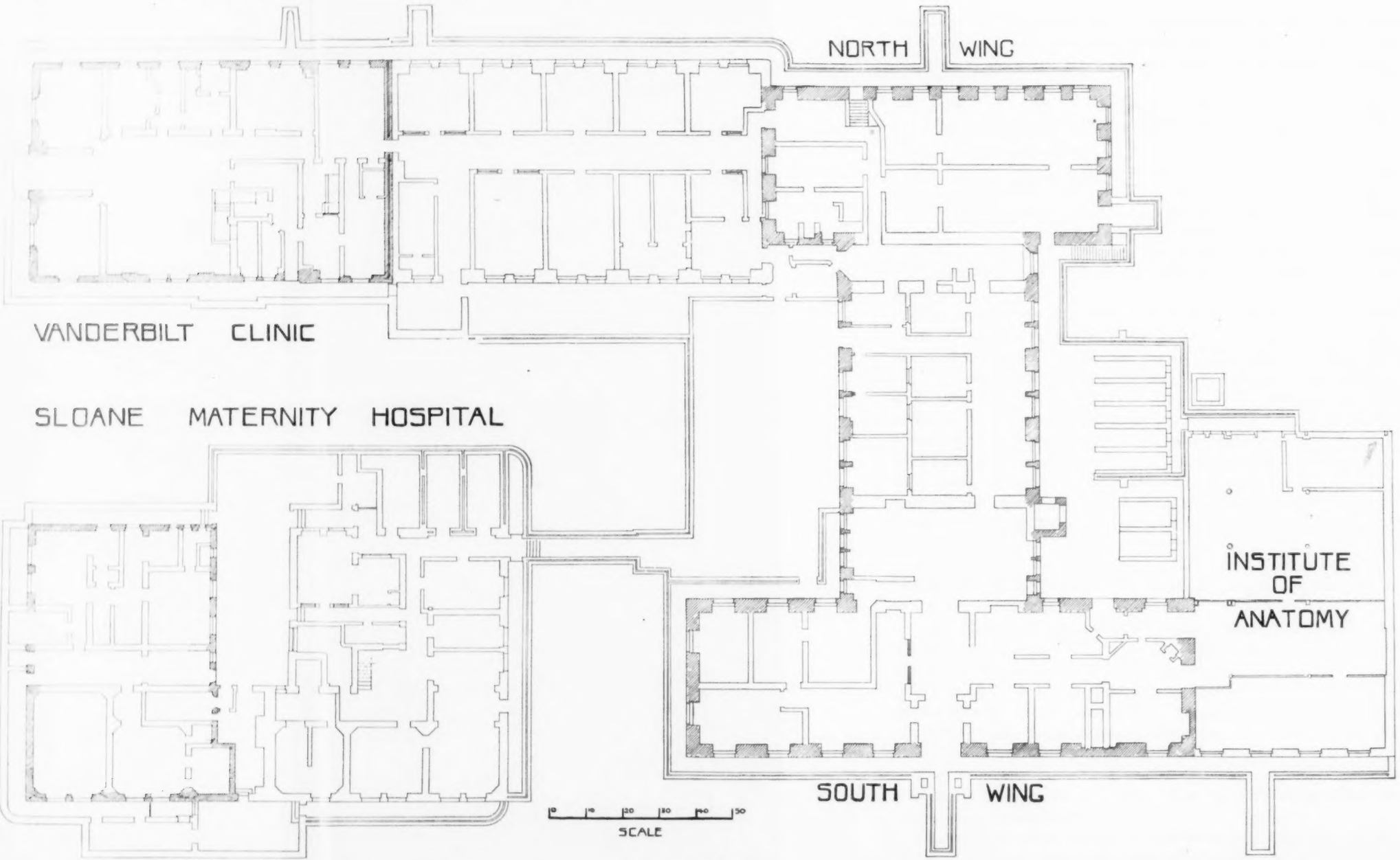

NORTH WING
VANDERBILT CLINIC
SLOANE MATERNITY HOSPITAL
INSTITUTE OF ANATOMY
SOUTH WING
SCALE

Fig. 235.

gave an opportunity not only to provide for their expressed needs but also to give to this body of workers the facilities of a modern hotel, office building, or factory, in so far as they could make use of such.

The result has not proved disappointing. The field of medical education has been so long a *terra incognita* to the average layman, wrapped in a seclusion that is most natural and fitting, no doubt, that one should not have felt surprise at finding this school,

FIG. 227.

in common with all others, keeping its dissecting material on ice or in pickle.

A 20-ton refrigerating machine was promptly installed and put to work, not only on this material in storage, but also in cooling the air for the great dissecting room in the top of the building.

It is interesting to note in this connection that when it was proposed to make this installation, the consensus of opinion among the members of the faculty was, that while it would, no doubt, be a good thing for anatomy, the other departments did not need it in their business.

That this attitude only showed conservatism in a profession that must be conservative, above all things, is evidenced by the fact that now artificial refrigeration is used everywhere in the school where it can be used, and has opened up fields for most fruitful investigation.

Its successful operation here has been followed by its adoption in several other medical schools and hospitals, both in this country and abroad.

The details of this part of the plant will be given further on. The subject has been referred to here merely to illustrate the wisdom of seeking to find new fields for old appliances.

The evident intention of the university trustees to advance the school as rapidly as possible was shown in the immediate addition of two stories to the north wing of the main building, which was immediately responded to by the original donors, who, in turn, provided funds for extensions that have doubled the original capacity of the buildings.

To enlarge the power plant and its accessories in a way that would entail the least waste of money already invested, and yet make a creditable engineering exhibit of the whole outfit from the standpoint of operating expenses, which ought to be the viewpoint of people who have to live on a fixed income, was a comparatively simple matter; although the putting together of a plant of the kind, which must run, at least in part, day and night all the year round, and which must serve a nervous and exacting population that cannot be expected to understand breakdowns or stoppages, demanded careful reflection.

Then there was the little matter of "indents," as McPhee would say, and the fact that a mistake will stare one in the face for a long time. When a college gets a thing, it usually has to keep it until it wears out or somebody gives it another.

The revamped plant is shown in plan.

The boilers are placed in a compartment by themselves, under what is in effect a court between the main building and the institute of anatomy, on the east side, with coal bunkers of 350 tons capacity opening into this compartment from the cellar of the north wing.

The space between the boiler room and the engine room contains the feed pumps and elevator pump, the 20-ton ice machine, and the heating and ventilating system for the original main building, air being delivered to the fans by down-takes

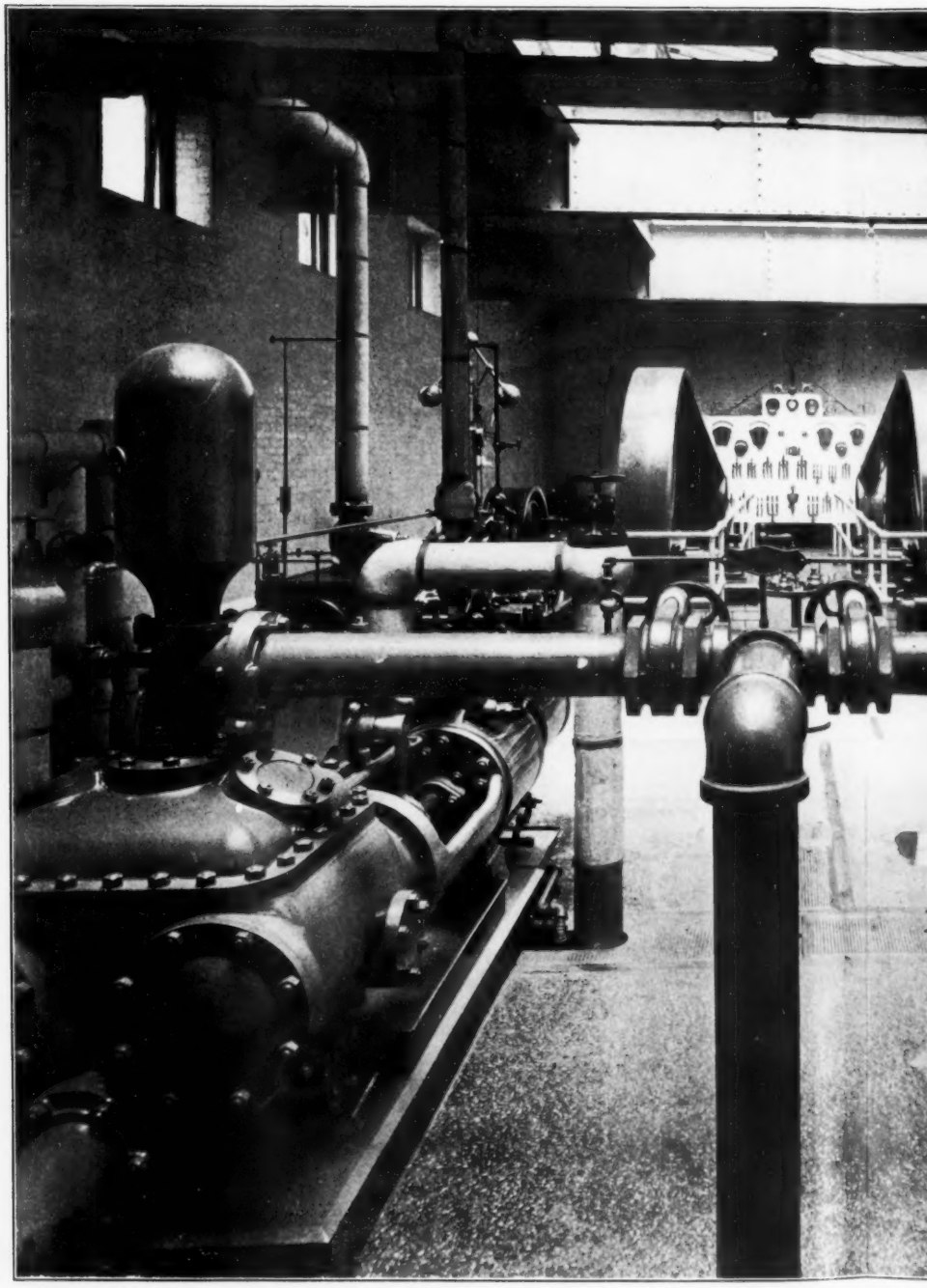
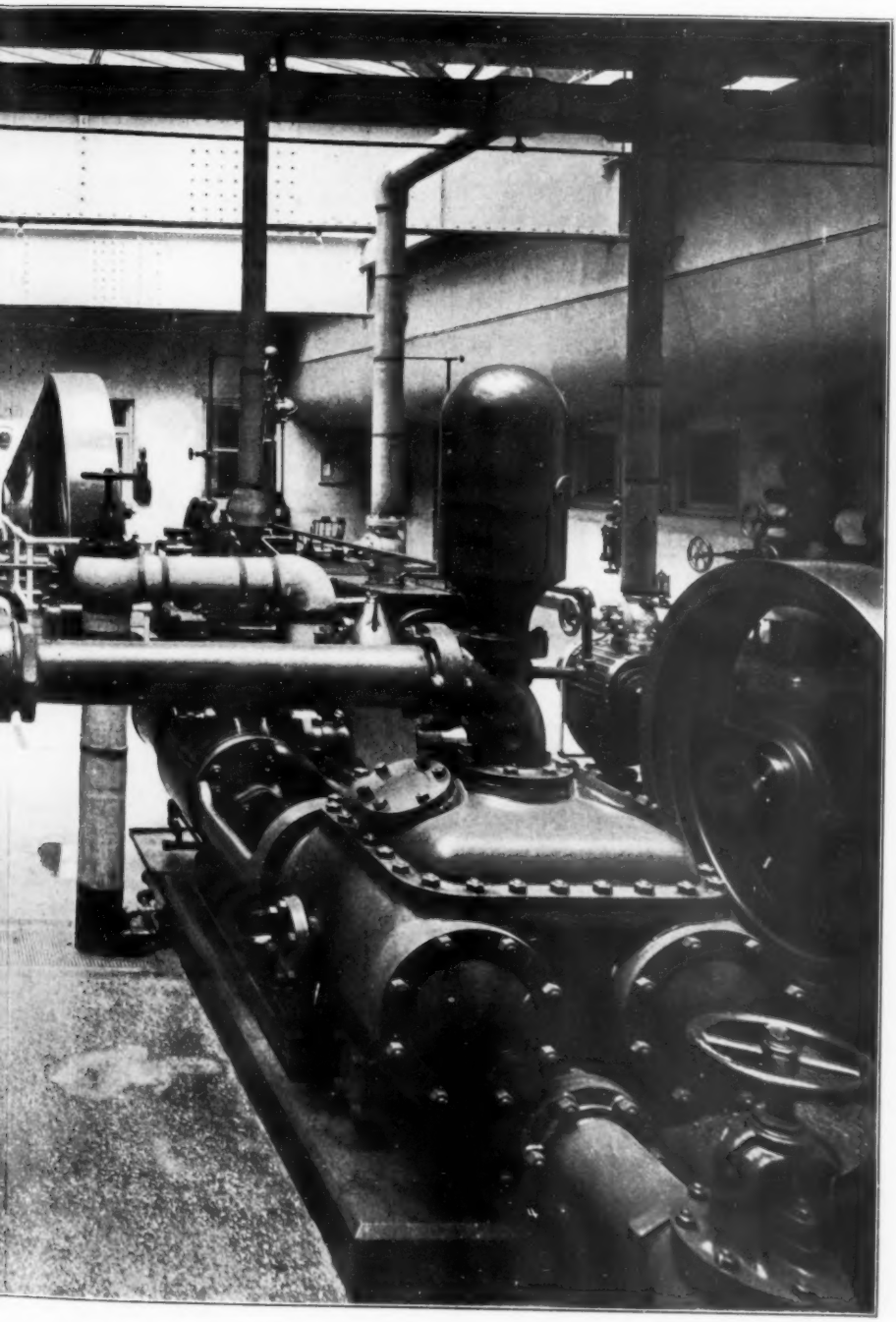


FIG. 228.



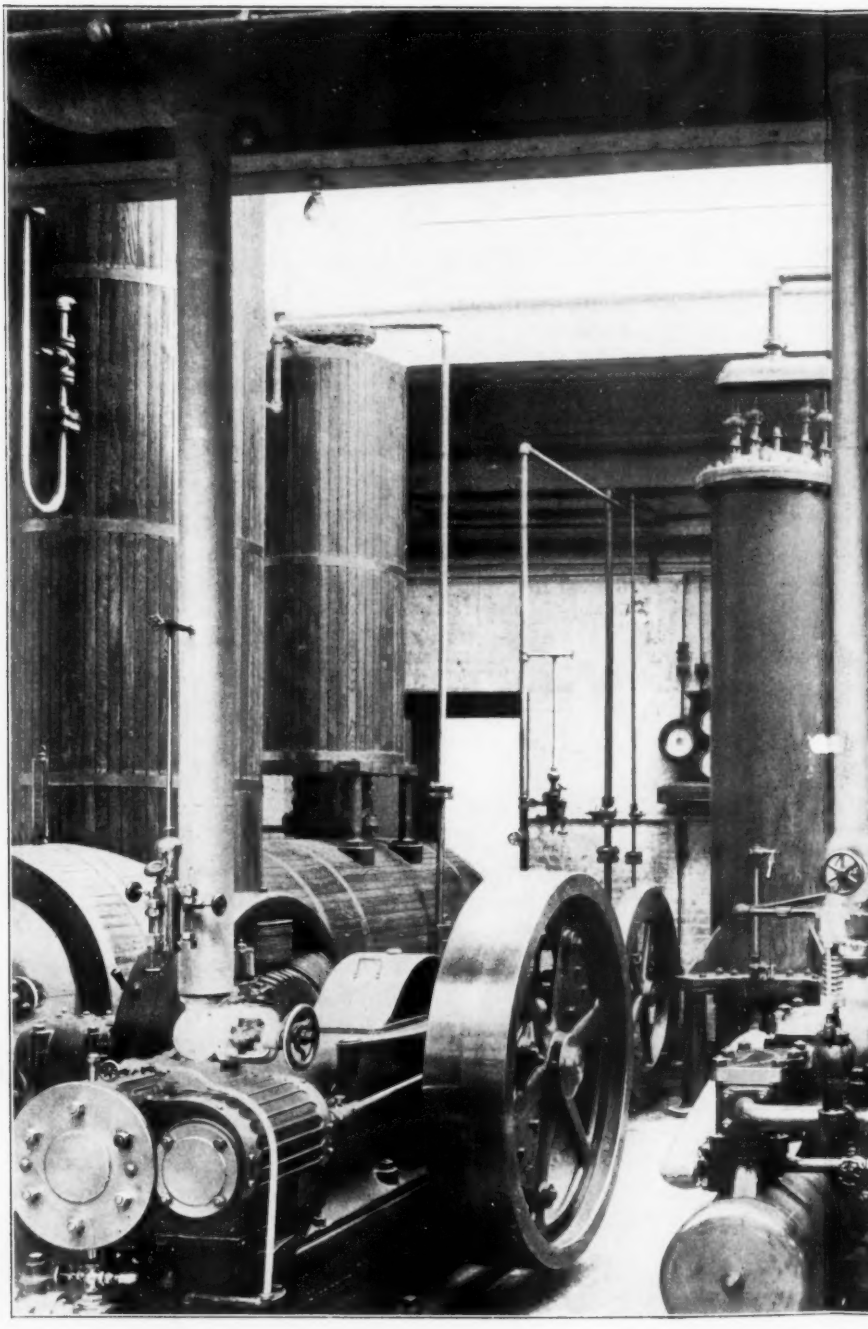


FIG. 229

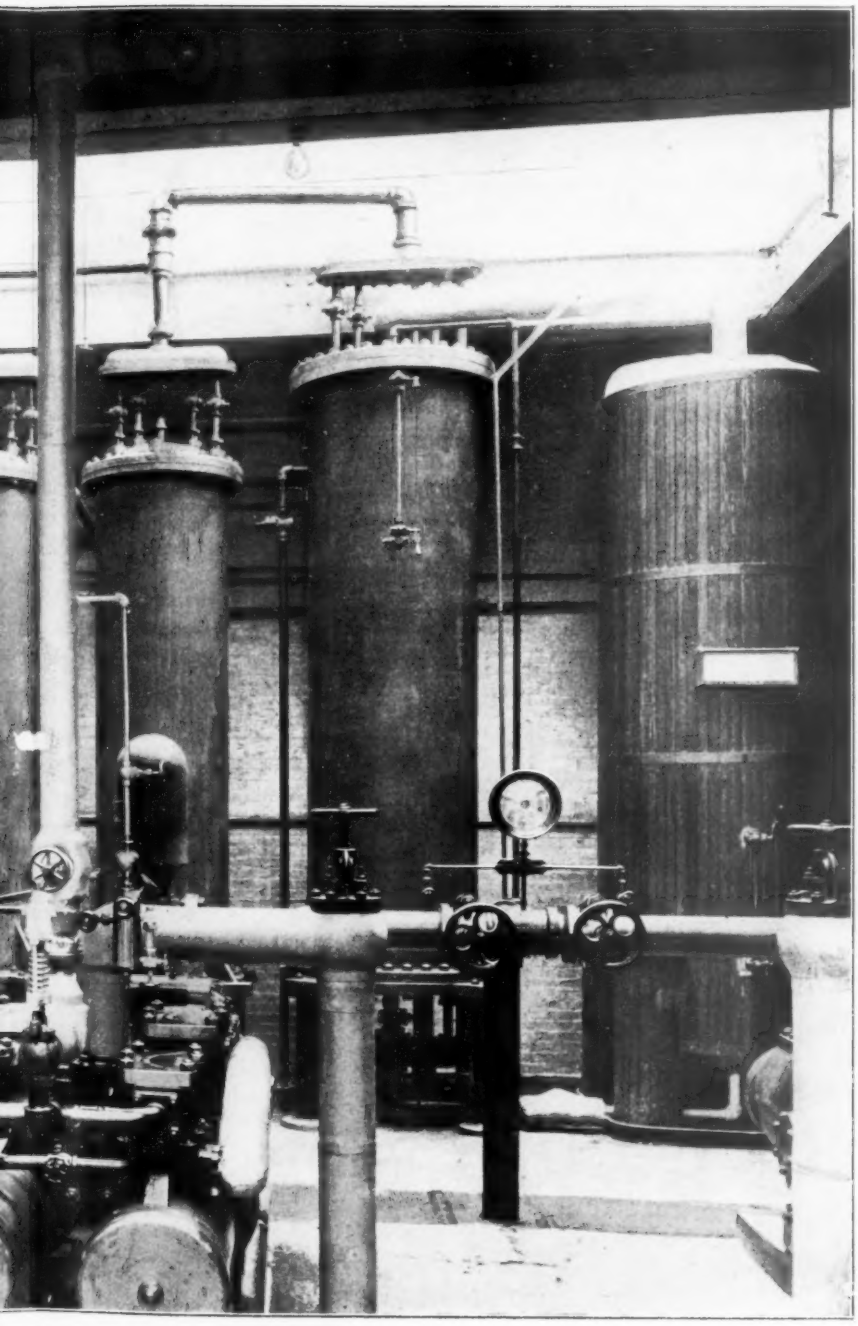


FIG. 229.

which are built up against the wall to the level of the third story, this arrangement being now used for all the fans in the entire group of buildings, to the end that we may serve air as free from dust as possible.

The engine room occupies a space similar to the boiler room, to the west of the main building. Like the boiler room, it is amply lighted by skylights, and is free from columns or obstructions of any kind.

This room contains the engines and generators, the water-service pumps and filters, and the 50-ton ice machine. The chief engineer, who has a desk in this room, can keep an eye on the water, light, and refrigerating service without leaving his chair.

Two views are here given of the engine room, one looking north toward the switchboard, the other looking south toward the ice machine. (Figs. 228 and 229.)

This room opens, it will be observed, north into the basement of the Vanderbilt Clinic, where the fans, heating stacks, and elevator machinery of that wing may be readily got at, and where are also located the repair shop and storage batteries, east, into the boiler room, south, into the Fifty-ninth Street wing, where the brine tanks and cold storage chambers are located, and west, through a tunnel into the basement of the Sloane Maternity Hospital. At the beginning of this tunnel may be noted a hatchway, for the taking in and out of machinery from the driveway overhead to the engine room below.

This point, by the way, is occasionally overlooked in the planning of buildings, as some of our members may know to their sorrow.

The plant is called on to serve the college proper in full operation during eight months of the year, and to a limited extent during four months, and the hospital and clinic the whole year round.

This exacting service knows no respite on Sundays or holidays, and requires for its operation a chief engineer, two assistants, one machinist, two oilers, two firemen, and two coal-passers who act also as helpers.

This force is divided to stand two twelve-hour watches.

On Sundays and holidays, the work being lighter, the force alternates on shore leave.

The wages paid are as high as the average in New York city,

and every man is given two weeks' holiday in the summer. Full pay is given for absence on sick leave, and the facilities of the clinic are extended to all the employees. The policy here outlined secures for the university a capable and willing service.

The Boilers.

There are five horizontal tubular boilers, 16 feet by 54 inches, rated at 60 horse-power each, carrying 40 pounds pressure, used for heating purposes, and two water-tube boilers of the Root pattern, rated at 150 horse-power each, having 108 4-inch tubes, 12 feet long, working at 110 pounds pressure, supplying steam for power.

All the boilers deliver their flue gases through an overhead breeching to a brick chimney 90 feet high, having an effective area of 14 square feet. This stack had a 36-inch cast-iron pipe set in it when first built to serve the original boilers, and when the new boilers were added this was abandoned and the annular space around the pipe used as a stack. Hobson's choice. We may some day go back to the iron pipe, and use a blower or an exhauster. The present method gives an ample draught, however, and the pipe is anchored into the brickwork in a way that defies extraction.

Anthracite buckwheat coal is used and is wheeled in large barrows from the bunkers. The ashes are wheeled in the same way to a hydraulic hoist, which in turn delivers to carts standing in the court, where the coal is weighed and delivered through a chute to the bunkers below.

Feed water is taken from two pumps, which in turn take water from the feed heater and the return and drip tanks.

Grease extractors are used throughout, and no difficulty has ever been experienced from the little oil that gets into the boilers.

The page from the log-book (see table facing) gives a week's run of the plant.

Ventilating and Heating System.

Air is forced into the buildings by a series of blowers located in the basements of the different buildings. The blowers which were put in first are belted to engines, the central group of five being driven by a single high-speed automatic engine; the

newer fans are motor driven, and it is hoped in time to have them all operated in this way.

Exhaust fans are used for accelerating the removal of foul air from laboratories where fumes are generated.

Heating is done on both the direct and indirect system. In the buildings first erected a system of twin ducts, one giving hot and one cold, was installed; mixing dampers set back of the register in each room being regulated to give the desired temperature, the cold air by-passing the heating stack set back of each fan.

In the newer buildings the incoming air is tempered as required, and radiators are placed under each window. The entire system was installed by Mr. William J. Baldwin, and has proved satisfactory in operation.

The most difficult point to ventilate, as may well be imagined, is the dissecting room.

This room contains seventy-six tables all told, and is the largest in use anywhere.

As first constructed, it extended the whole length of the south wing and was lighted by a series of large skylights of the ordinary pattern. During the winter months there was no trouble in controlling the temperature in the room, but when the sun was high, in the fall and spring, the skylights acted like the roof of a greenhouse, and heavy canvas awnings had to be rigged over their southern facets.

When the L was added to the south wing and its top story was thrown into the dissecting room as an enlargement, we borrowed a leaf from the book of experience of those who operate factories, and put a weaving-shed roof on, to give north light only.

Fresh air is sprinkled into the dissecting-room through the perforated cornice extending all the way round, and the foul air escapes through registers located near the floor. In warm weather registers set at the base of the skylights and in the ends of the weaving sheds are opened and the system reversed.

When the first ice machine was put in we set up a coil of 1½-inch brine pipe, containing over a mile of pipe, in the settling chamber of the ventilating fan that supplied this room, and when the temperature of the outside air rose above 70° turned on the brine circulation and held the temperature down to the point named.

This scheme, though effective, proved very wasteful in operation, as the air was reheated in coming up through the outside building walls.

When the new wing was added we absorbed the heat directly by running brine coils along the ends of the weaving sheds, and we find this way more efficient. While the room would hardly pass muster as a sanitarium, the air is remarkably good at all times.

As the arrangement described is the first of its kind, a photograph, which has been reproduced in Fig. 230, may prove interesting.

Attention is called to the drip trough under the coils.

Perhaps the chief benefit derived from the use of the cooling coils is the drying effect produced on the incoming air. On humid days a considerable stream of water trickles away in the troughs.

It is a well-known fact that in dry climates, even where a high temperature obtains, meat is cured in the open air. Moisture in the air induces decomposition and the consequent generation of offensive gases.

Brine coils are also used for cooling the air and extracting its moisture in sultry weather, in the ventilating system of the Maternity Hospital. In this case the coils are distributed in the down-take air shaft, and they are staggered in order to baffle the current and at the same time render the coils easy of access for inspection and painting. A vertical section of the shaft is shown in Fig. 231.

Refrigerating Plant.

This consists of two Polar absorption machines, one of 50 tons capacity, the other rated at 20 tons.

The choice of this system was made on account of the comparative simplicity of its operation and its freedom from noise and vibration, in a place where these points are all-important.

The large machine is used during the warm months and the smaller for the light service demanded during cool weather.

Steam is supplied to the generators at 45 pounds pressure, through the medium of a reducing valve.

The 50-ton machine requires a working charge of 700 pounds of anhydrous and 5,000 pounds of aqua ammonia, and the 20-ton

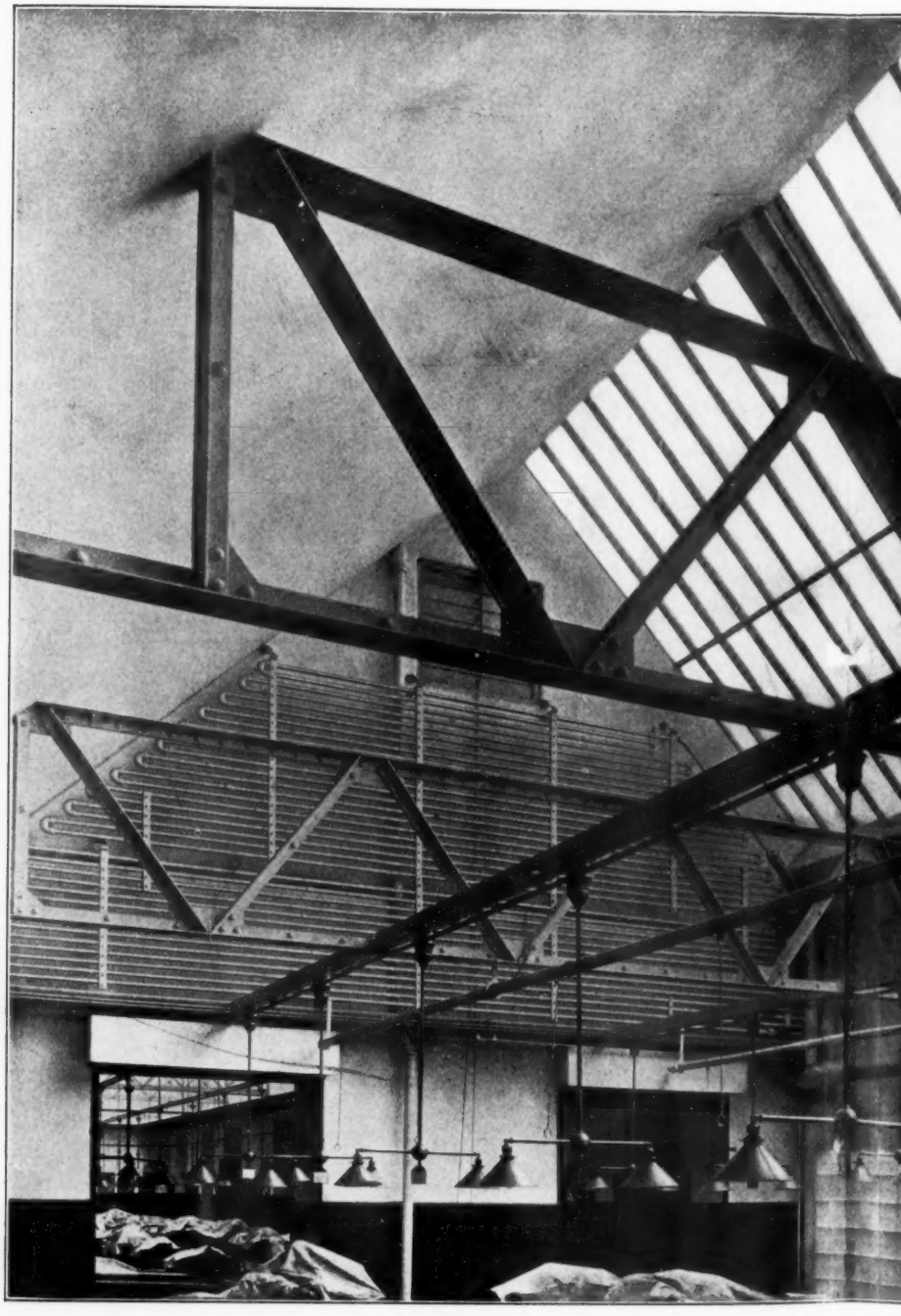


FIG. 230.

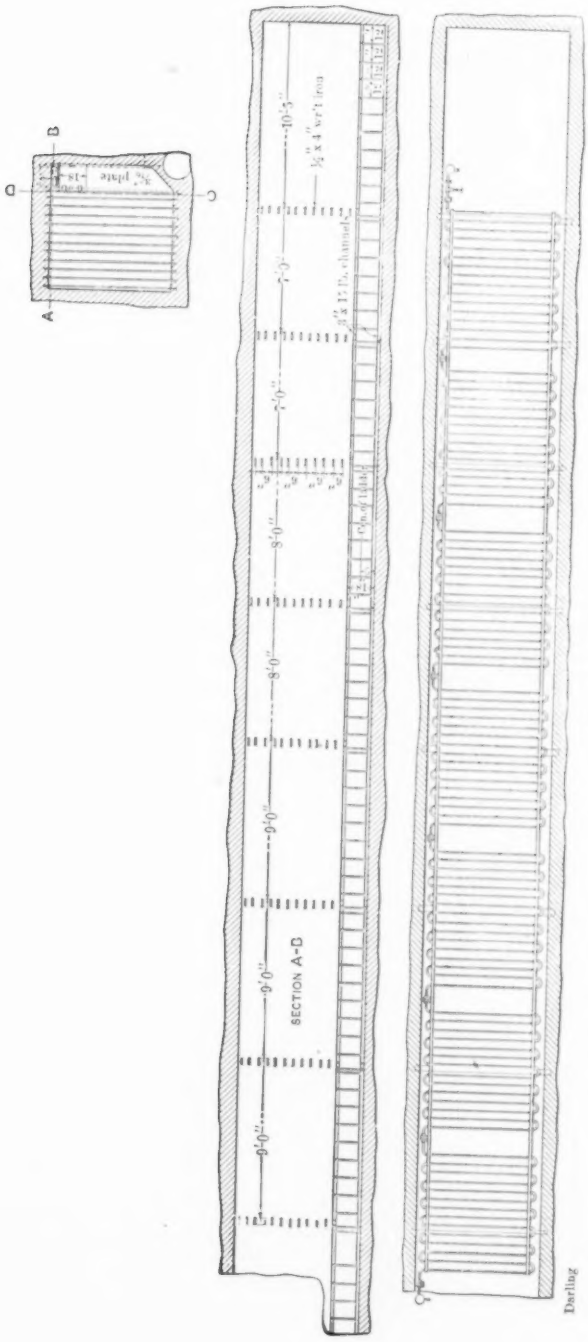


FIG. 231.

machine 300 pounds of anhydrous and 2,100 pounds of aqua ammonia.

Condensation is provided for by the circulation of the house-service water, a large amount of water fortunately being used throughout the buildings. A brave attempt was made to get well water for this work, on account of its uniformly low temperature, but a hole 8 inches in diameter, driven down 250 feet in the seamy gnarled rock formation on which New York stands, barely yielded enough moisture to wet the drills.

The driller offered to go on till he struck the usual possibilities, after the manner of his kind, at $5 per foot, perhaps; but we tarried there. Some day we may take counsel of our courage and make another stab at it. Others have succeeded, and this is not commonly counted a dry town.

Chloride of calcium has been used from the first in making up the brine, of which there is 50 tons in circulation. It does not corrode the pipes in the first place, does not deposit at zero temperature in the second place, and it costs little, if any, more than common salt.

Particular attention is called to what may seem the abnormal size of the brine tanks.

Those who are familiar with New York weather will remember that it blows hot and it blows cold in a somewhat trying manner.

The large tanks act as storage batteries, so to speak, and take care of the peaks of the load, allowing the ice machines to follow the even tenor of their way at a normal working rate.

During the hot weather the machines are run day and night, catching up at night on the heavy drafts made on them by the heat of the day.

During cold weather the machines are shut down for as long as forty-eight hours at a time, the great body of cold brine rising only a few degrees in temperature during that time, the only sign of life about the plant being the slow-moving brine pump.

The work they are called on to perform is of a varied character. The cooling of the ventilating air has already been alluded to. There is in addition the operation of two cold-storage rooms for the department of anatomy, these rooms ranging from 10 degrees to 20 degrees Fahr.; a cold-storage room for pathological material, kept at about 30 degrees Fahr.; a series of refrigerator compartments for bacteriological cultures, ranging from

10 degrees to 60 degrees Fahr. as required; three cold-storage rooms and six refrigerators for keeping vegetables, meat, milk, and other food supplies in bulk and in detail in the hospital. Then in addition to these is the making of the ice needed in the whole establishment, averaging about 250 pounds per day.

It would be a somewhat difficult and intricate matter to keep a tally sheet on all this work as an index to its efficiency, although both machines have been carefully tested and found able to exceed their rated capacity on the coal and water consumption stated in the specifications.

What is more important, from the operator's standpoint, is the knowledge that they have furnished continuous service without a serious breakdown since the first one was installed, six years ago.

The form of log kept for the ice machines is here shown:

TABLE J.

COLUMBIA COLLEGE.—MEDICAL DEPARTMENT.

DAILY RECORD OF REFRIGERATING APPARATUS.

Time.	Steam Pressures.		Ammonia Pressures.			Brine Temperature. Cooler.		Water Temperature.		Remarks.
	Boiler.	Generator.	Generator.	Cooler.	Absorber.	Inlet.	Outlet.	Cond'r. Inlet.	Absorb Outlet.	
A.M.										
1....	60	45	80	15	14	24	24	41	98	
2....	65	45	80	14	13	22	16	41	99	
3....	60	45	80	13	12	20	14	41	98	
4....	60	45	80	12	11	19	13	41	88	
5....	65	45	80	11	10	18	12	41	98	
6....	65	45	80	11	10	17	11	41	99	
7....	65	45	80	11	10	16	10	42	100	
8....	65	45	85	10	9	15	9	42	98	
9....	65	45	90	10	9	14	8	42	98	
10....	65	45	90	9	8	13	7	42	98	
11 ..	65	45	90	9	8	12	6	42	98	
12....	65	45	90	8	7	12	6	42	98	
P.M.										
1....	65	45	90	8	7	10	5	42	98	
2....	65	45	90	8	7	10	5	42	99	
3....	65	45	85	8	7	9	4	42	99	
4....	65	45	85	7	6	9	4	42	99	
5 ..	65	45	85	7	6	9	4	42	100	
6 ...	65	45	85	7	6	9	4	42	98	
7 ...	65	45	80	6	5	8	2	43	99	
8....	65	45	80	6	5	7	1	43	99	
9....	60	45	80	6	5	6	0	43	99	
10....	60	45	80	6	5	5	0	43	99	
11....	60	45	80	6	5	4	0	43	100	
12....	60	45	80	5	4	3	0	43	100	

Dated January 24, 1898.

Elevators.

There are two combination freight and passenger hydraulic elevators and a hydraulic ash lift in the main building and the institute of anatomy wing, served by a 14 by 10 by 12 duplex pump located in the engine room.

The elevators are of the ordinary Otis type. The ash lift was specially designed by the Otis Company, and is worthy of notice on account of its extreme simplicity, quick motion, and ease of handling. It is shown in the accompanying cut (Fig. 232).

The ash buckets are lifted directly from the level of the boiler room floor to the tail of the cart, to be dumped, the load rising at double the speed of the ram.

In the hospital and clinic there are two electric passenger elevators and two electric freight elevators, one of which delivers supplies to the hospital kitchen, which is located on the sixth floor. An electric dumb-waiter operates between the kitchen and the dining rooms of the wards on the floors below.

This arrangement of wards, by the way, renders isolation in case of sepsis a comparatively easy matter. It also makes easy the disposition of soiled clothing, etc., by means of a large copper-lined chute with inlets at each floor and delivering directly into the laundry in the basement.

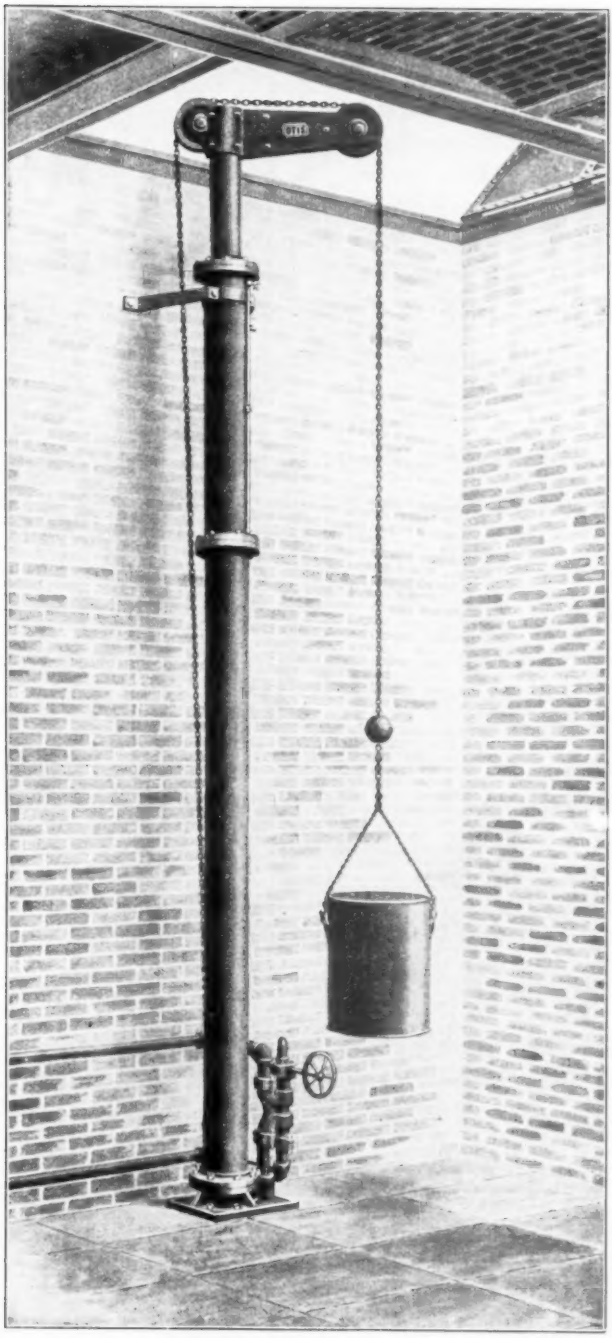
Auxiliary Apparatus.

Under this heading may be mentioned: The steam sterilizers for the operating rooms, in which all the paraphernalia of a modern surgical operation is sterilized.

The air compressor for supplying air under pressure for operations on the nose and throat, the air having been previously freed from dust by passing through a filter chamber filled with cotton-wool.

The culture ovens and sterilizers used in the bacteriological preparation room, some of which are shown in Fig. 233.

The principal feature of this outfit is the sterilizer, which we designed for the preparation of serum on a large scale. The test tubes, filled with serum and corked with cotton-wool, are placed in wire baskets, which in turn are deposited on wire trays resting on lugs cast on the inner wall of the sterilizer, which is a simple steam-jacketed kettle. The heavy lid of the kettle is counterbalanced and comes down on a thick rubber gasket, made



44 FIG. 232.

steam-tight by means of hinge bolts. Steam is then turned either directly into the kettle or into the jacket as may be required; a trap returns the condensation to the drip tank in the boiler room.

Some idea of the extent to which mechanical engineering serves medical instruction may be given when we state that the physiological laboratory uses 100 feet of line shafting in its business and keeps a first-class machinist busy making and setting up apparatus all the time.

In the department of anatomy there is, in addition to the

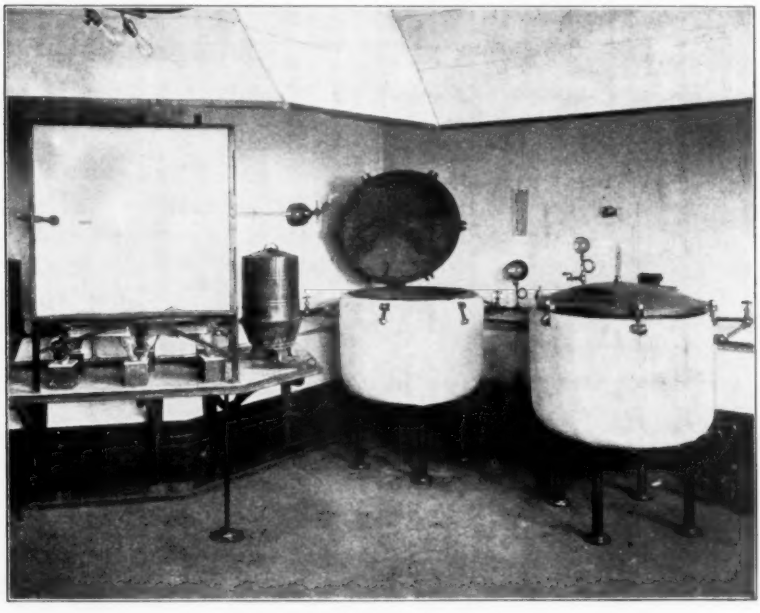


FIG. 233.

refrigerating equipment already mentioned, a macerating room fully equipped with steam and corrosion vats and a reverberatory cremating furnace in which a fuel-oil blast is used. The work in this room covers the preparation of material for the museum of human and comparative anatomy, and the apparatus used must be of sufficient capacity to take care of the largest mammals.

Several elephants have emerged from its processes in the form of gleaming skeletons.

The articulating and mounting of bones are done in a well-

equipped little shop by a machinist who has been trained here in this line, and who, like his mate in Physiology, was recruited from a neighboring iron-works.

An addition to this equipment is contemplated in the shape of an outfit for making frozen sections of entire animals, which will be fastened down on a carriage and run through a band saw.

In concluding this part of the paper, it is only fair to state the impression of the writer, gained in an extended period of service in this direction, that there is now no profession more keenly alert to the advances made along what it perceives are parallel lines in other fields of human effort than the medical profession.

The New Site.

When, in 1891, the trustees came to the conclusion that Columbia University could not very well grow with a roaring railroad on one side and prohibitive real-estate values on the other three sides, they appointed a committee on site to report what had best be done under the circumstances.

Immediately there began to come advice to the committee from all sides, the alumni, the press of the city, and friends who take an interest in Columbia's welfare.

Some thought that the university should be taken bodily out of the city and be planted in the country where ground was cheap and shade was plenty.

Others took the view that the college or academic department might be given a sylvan retreat, but that the professional schools had best be kept in close touch with city life.

The committee, after patiently hearing all sides, took the safe middle ground and recommended to the trustees the purchase of a site which, while it lay on the exact geographic equator of the city, was in a region of parks and broad avenues and was of sufficient extent to ensure ample breathing space and the opportunity for great expansion.

The name of this region was originally Bloomingdale, an anglicized form of Blomendael, a suburb of Haarlem, which is in turn a suburb of the old city of Amsterdam—names most appropriately given by the Dutch pioneers to their then new settlements, and pity it is that, in the ruthless growth of the city, these old names bid fair to be merged into mere numbered wards and districts so meaningless when compared with the

historic regions of Rome, which have lasted down the centuries and almost tell its history.

One of the battles of the Revolution was fought on this ground, and during the war of 1812 a line of earthworks was thrown up across it to defend the city from the north.

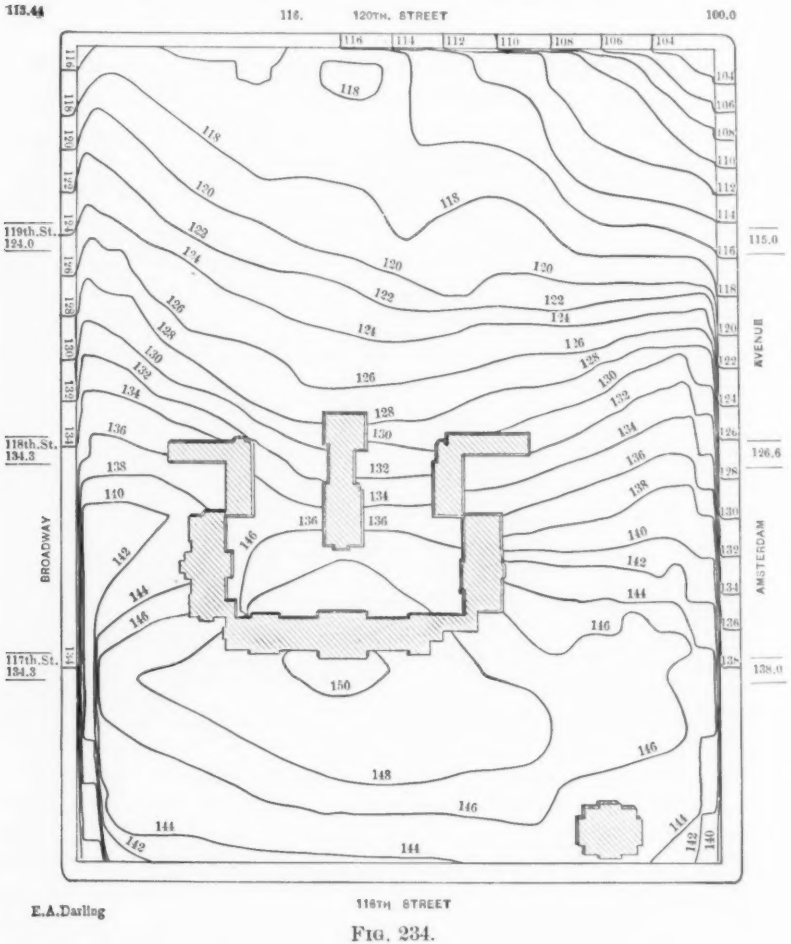


FIG. 234.

In 1825 it was purchased by the New York Hospital for the establishment of an insane asylum, and the name became almost a byword in the land.

The purchase of the site, at a cost of two million dollars, was completed.

It comprised a plot 16½ acres in extent lying between Amsterdam Avenue and Broadway on the east and west and between One Hundred and Sixteenth Street and One Hundred and Twentieth Street south and north.

The trustees immediately entered upon a campaign of deliberate and careful study of the problem before them. This campaign covered a period of three years, and not one moment of time was wasted. It meant simply "Be sure you are right, *then* go ahead."

It meant the development of a building plan that would be in effect a written constitution for the growth of the university for at least two generations to come; a plan that would combine architectural dignity and symmetry with a mobility that would meet the constantly changing needs of university life and would provide for the ultimate use of every foot of ground for buildings that could be fairly demanded by succeeding generations of teachers and students.

Architecture is perhaps the oldest art in the world, and the development of such a plan did not call for invention on the part of the advisory commission of architects whom the trustees called to their aid.

Building schemes in every variety of form that have required ages for their realization exist in the cities of Europe, and a patient process of selection and adaptation will yield sure and safe results.

After a careful survey, which included a series of borings, a topographical relief map of the site was made (Fig. 234), and comparative plans for the entire development were submitted by the architects, who composed the advisory commission.

The plan from Messrs. McKim, Mead & White proved most acceptable to the trustees, and that firm was retained to carry forward the work.

At first it was thought that the substantial group of buildings formerly used by the asylum, and representing an investment of $900,000, might be worked into the plan, in order to lighten the financial burden that began to loom large in the development, but it soon became apparent that they would have to be ignored, because they prevented any scientific development of the ground as a whole. Accordingly only such as did not immediately interfere with the new buildings could be made to serve temporary purposes.

A patient process of evolution and reduction finally brought the plan to the form finally adopted, which is shown in Fig. 236, together with profiles running north and south on the major axis, and east and west on the minor axis, of the group (Figs. 237 and 238).

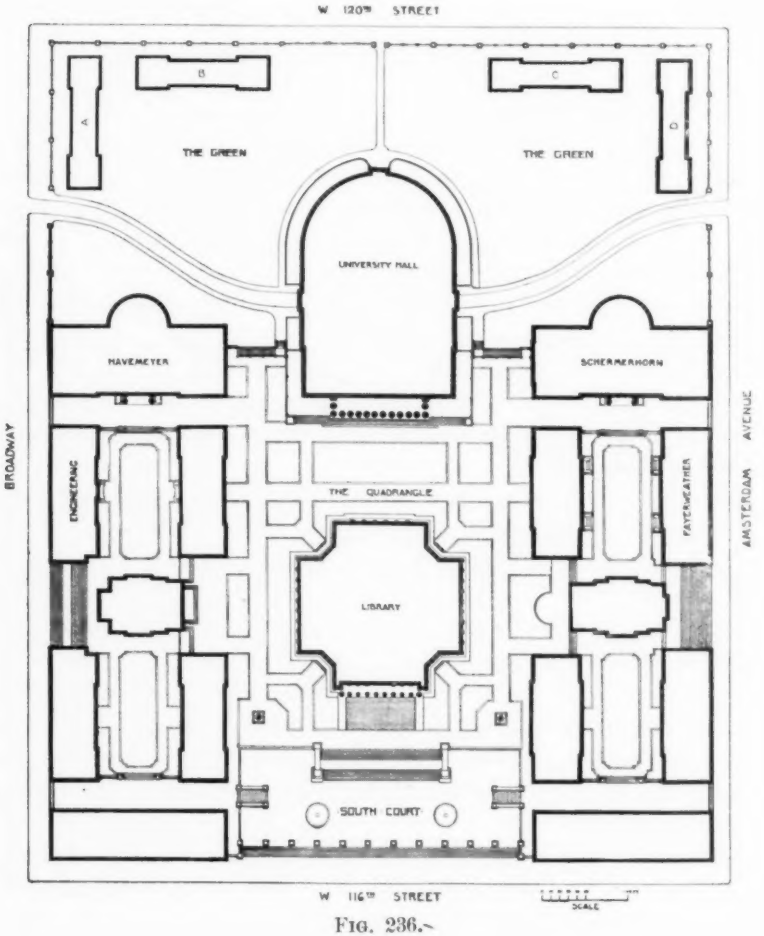


FIG. 236.—

The plan called for the creation of a great platform, 750 by 575 feet, comprising the southern portion of the plot, taking the highest natural level, which was 150 feet above the Hudson River and from 12 to 25 feet above the surrounding streets.

On the centre line of this platform the library building (Fig.

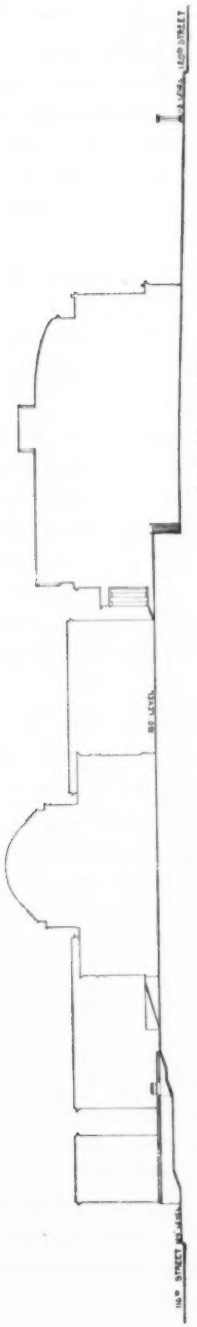


FIG. 287.

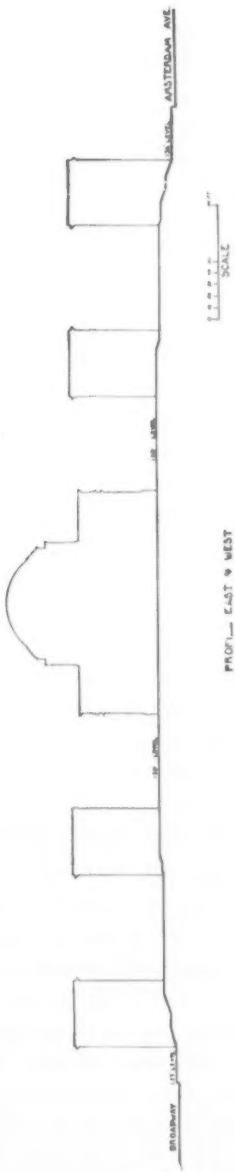


FIG. 288.

252) was to form the pivot. Directly back of the library, which faced south, University Hall, containing the refectory, academic theatre, gymnasium, baths, and power plant, was to be placed; and flanking the library on the east and west were four groups of educational buildings, forming as many quadrangles.

The basements of the buildings on the outer line of the platform are entirely above ground, and the breaks between the buildings are connected by retaining walls.

The basements of the inner line of buildings facing the library are entirely below ground.

The buildings have a common height of 69 feet from water-table to cornice, and the height of stories is the same in all buildings except the library and University Hall.

North of the platform the natural level of the ground conformed nearly to the level of the surrounding streets, and a fine grove of trees that occupied this portion suggested the creation of a college green, to be surrounded by an ornamental fence and interlaced with drives and walks.

When the plan had been decided on, two institutions, which are now a part of the university system, the Teachers' College and Barnard College, purchased sites directly facing the green, to the north and west.

It has been subsequently decided to place four dormitories around the green for the accommodation of students. These are shown on the plan, lettered A, B, C, and D.

The Italian Renaissance was the style decided on for the entire group of buildings, as being best adapted for an urban composition, giving as it does the greatest opportunity for securing abundant light and that flexibility of treatment demanded by the diverse and constantly changing phases of university life.

Uniformity and symmetry in plan and elevation called for harmony in color and texture.

Granite was the foundation to start with, and it was accordingly adopted for all basements, retaining walls, steps, and balustrades, and the masonry of the fences and gates.

It must needs be of a warm tone to harmonize with the limestone of which the axial buildings were to be constructed, and with the red brick and limestone of the outlying buildings.

All pavements, both inside and outside the grounds, are of red brick with Joliet stone borders. The entire color scheme is

thus kept warm in tone, an effect that is appreciated in buildings which are looked at most in winter, and that harmonizes well with the green of the grass and trees in summer.

With a view to securing the lowest possible insurance rates, as well as the desire to avoid the inevitable embarrassments and delays caused by fire, it was decided to use throughout the most approved form of fireproof construction. All trusses, girders, and beams must therefore be of steel, all columns either of steel or iron, and all floors, partitions, and roofs of hollow brick or tile. The columns and girders should be protected from fire by tile or expanded metal lath, plastered.

Stairways either entirely of stone or of iron and stone.

To make the roofs, rain leaders, gutters, and skylight framing as durable as possible, it was decided to use copper for these purposes.

The plumbing, heating, and ventilating apparatus, and electric wiring were to be of the very best known in these branches, and plate glass was to be used for all windows. All other appointments were to be of the grade used in the best office-building construction.

The decision to build along these lines and use materials of the grades noted meant the setting of a financial pace that would be difficult to follow. While it showed simply a determination on the part of the trustees to live up to the often preached but too seldom practised maxim "The best is always the cheapest in the end," it enlarged the financial problem in a way that was well-nigh appalling, for the money needed must either be secured by gift or borrowed on bond and mortgage.

That the courage of the men who administered this trust—for it was real courage—was not misplaced is evidenced by the fact that four out of the six buildings already erected have been provided through the munificence of men and women who believe in giving a university the best buildings the art of the times affords.

It is the privilege of one who is perhaps better qualified than any other, through a knowledge of the facts, to state here, in this connection, that each of these structures was built absolutely without restriction from the donor as to shape, style, texture, or even cost, within broad limits.

The commissions simply read, "Make the best building possible for the purpose intended"

These broad-minded benefactors gave their confidence with their money, and their confidence has been met with a spirit on the part of those to whom it was given that would not knowingly waste one penny of the money.

New York has not yet satisfied in full the demands made on it by its old university, but it will, and that right handsomely.

Drainage.

The first work necessary after the preparation of the preliminary plans—what may be called the reconnaissance, the establishment of grades and levels—was the laying out of a comprehensive system of drainage.

This was entrusted to Messrs. Waring, Chapman & Farquhar, a firm of civil engineers with ample experience along this line, and the result of their labors is shown in the plan exhibited in Fig. 239.

The sewers in the adjoining streets had not been built, and the coöperation of the Department of Public Works was sought and obtained, with the result that the city permitted the university to construct these sewers in connection with its own work, subject to the approval of the municipal authorities.

The city government also agreed to the establishment of street levels on the north and south sides of the plot on lines that would harmonize with the university plan. This was a most important concession, as the grades ran in opposite directions on each of these streets, and the effect of the beautiful entrance court on One Hundred and Sixteenth Street would have been marred by an abrupt grade at this point.

The main sewers are of the regulation oval section of hard brick laid in Portland cement mortar. Branch lines and connections are of heavy vitrified pipe. It will be noted on the plan that the flow is divided north and south from a centre line running through the library building. The undeveloped part of the plateau, together with the old buildings, which are serving temporary purposes, had to be taken care of; and in order to make them fit the plan it was necessary to reverse the drainage from them and send it into the new system, and a very tedious and expensive operation it proved, too.

The north and south trunk lines from the interior court had

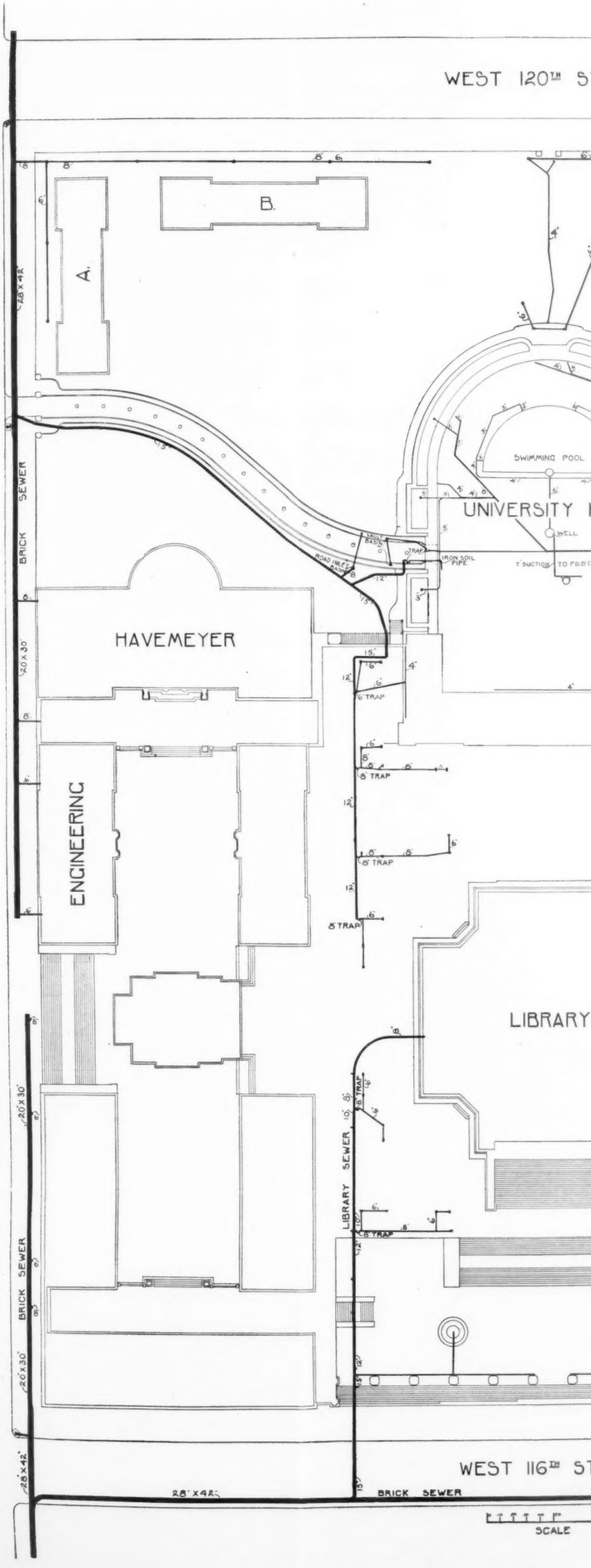

WEST 120TH ST
B.
A.
28"X42"
BRICK SEWER
SWIMMING POOL
UNIVERSITY
WELL
VAULT BASIN
TRAP
IRON SOIL PIPE
SUCTION
TO FEED
ROAD INLET BASIN
HAVEMEYER
20"X30"
ENGINEERING
6" TRAP
8" TRAP
LIBRARY
LIBRARY SEWER
BROADWAY
WEST 116TH ST
SCALE

FIG. 239.

STREET

C

D

TRAP

28 x 42

8' INSPECTION

8' INSPECTION

15"

8' INSPECTION

BRICK SEWER

AVENUE

AMSTERDAM

POOL

Y HALL

ELL

O FEED PUMPS

SOIL PIPE

MAN HOLE

TRAP

VAULT BASIN

12"

SCHERMERHORN

FAYERWEATHER

20' x 30'

6" TRAP

8" TRAP

8" TRAP

RY

8" TRAP

LIBRARY SEWER

STREET

28' x 42'

28' x 42'

39.

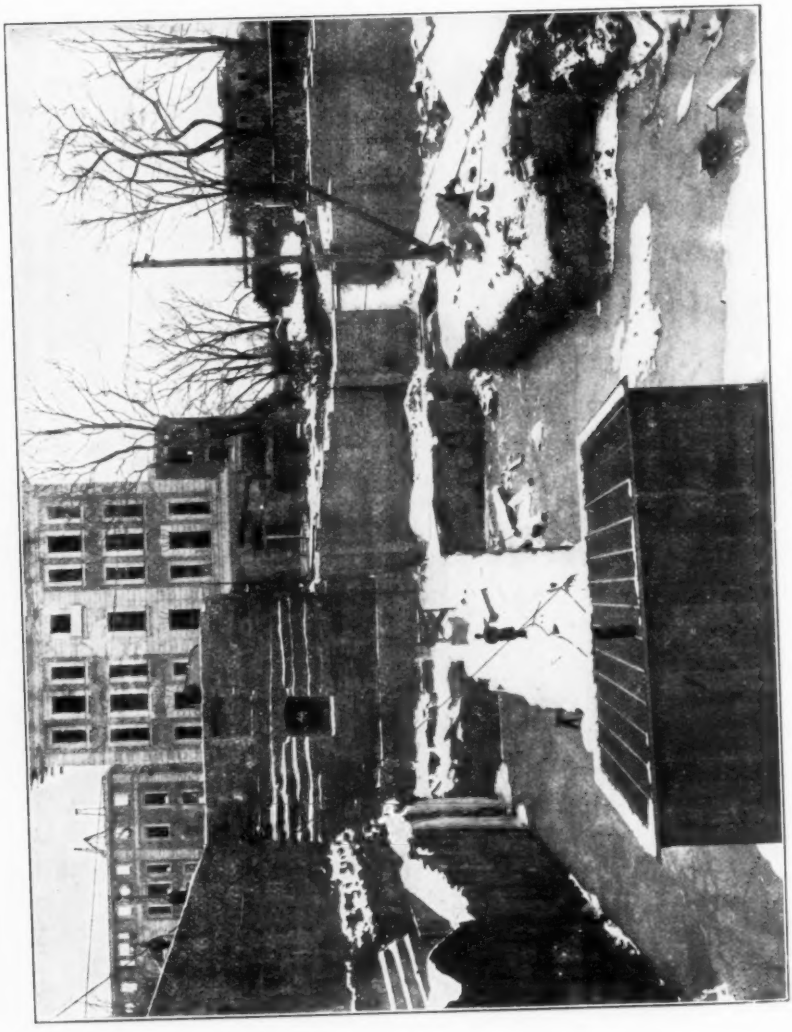


FIG. 240

The Power Plant.

After the general layout had been determined, the question of levels demanded most careful consideration. There was involved in this question not only the drainage system and the probable cost of rock excavation on a great scale, but the important point of articulation for the supply of heat, light, water, power, etc., to the various buildings and for the transfer of freight to and from the library and the scientific departments.

The location and extent of the power plant had been from the first kept well to the fore by the writer, who, it will be readily understood, desired that the engineering of the enterprise should at least be in keeping with the architecture.

In all educational institutions there is the constant temptation to kill two birds with one stone by making the power plant illustrate the various phases of machine design. When you try this you generally miss both birds, and produce a menagerie of misfits.

Whatever value a power plant may have as an educational exhibit must depend solely on how well it serves the purpose for which it was designed.

Combination tools are usually found in the hands of amateur mechanics.

Realizing that a power house would be at the best a disturbing element in the midst of academic repose, I proposed to build it down on the bank of the Hudson River, 1,500 feet west in an air line, sinking a shaft from the centre of the university site and connecting from this a tunnel that would run under the Riverside Park and the New York Central tracks to the power house built out over the water on piers. Through the tunnel and shaft were to run pipes and cables for the supply of steam and electricity, and a light railway and elevator to carry freight.

The apparent exhibition of force at the university would then have been a bank of valves and a switchboard in the basement of the central building.

This scheme, which would have made possible the transfer of coal directly from the boats to bins over the boilers, and the use of condensing engines, received the sanction of the trustees and the consent of the city and the railroad authorities.

It was blocked by legislation introduced at Albany by an organization of adjacent property owners, who, while they had

in view the preservation of the water front of the park, could not understand that a constant stream of coal and ash carts up the hill, with a smokestack on the top of it, would be far more unsightly than a solid, handsome structure with a low stack for forced draft, under the hill. These men, no doubt, meant well, but the immediate foreground of the view from Grant's Tomb is at this moment filled by a great ungainly, unlovely, garbage dump that the city had to establish at this point, and another is projected almost within sight to the south of it. We had perforce to retreat within ourselves and cut a great cavern in the rock to hold our boilers and engines, and mask our chimneys in the building overhead. Fortunately this is of sufficient bulk to hide a multitude of sins.

In place of a power house, the site selected was used for the erection of a fine boat house for the university crews, and we proceeded with the plan for a power house in the centre of the site.

In order to determine at this stage of the proceedings the amount of space that would be required for boilers and machinery to operate the entire group of buildings, the writer had to draw on his imagination to a very considerable extent, take all he could get, and let it go at that.

It seemed to the writer that what might be called the "architectural" part of the problem in hand was relatively more important than that which related to the types and sizes of boilers and machinery, for the reason that bricks and mortar, in the shape of buildings, coal vaults, tunnels, flues, etc., would probably remain indefinitely as fixtures, while the plant was liable to change from time to time as parts of it wore out or were replaced by more efficient units, as the state of the art advanced.

The first consideration, then, was to secure the best physical conditions obtainable in the allotment and disposition of space in the power house and its accessories, having in mind the fact that these dispositions, once made, would become fixed, and a mistake or misconception along this line would remain a constant source of embarrassment, not only to ourselves but to those who would come after us.

As the power plant had to be in the same building with the gymnasium, baths, dining hall, and theatre, and had to hold its own in a campaign of planning and counterplanning which kept the building on the drawing boards of the architects for a

year and a half, the final issue from the conflict with a room 165 by 100 feet, 33 feet high in the clear, was not bad.

In the meantime the plans for the educational buildings had been carried to completion, contracts were let, and the buildings put under construction.

Mr. Alfred R. Wolff had been retained to prepare plans and specifications for the heating and ventilating apparatus, and Mr. C. O. Mailloux to do similar service in electrical wiring. These gentlemen were requested to do the best they knew how, in their respective lines, without restriction, and that they rose to the occasion has been demonstrated by the outcome of their endeavors.

It was agreed that a steam pressure of 110 pounds should be maintained throughout the distributing mains to the point where it met the reducing valves of the several heating systems, this pressure being taken as the limit for the single-expansion engines which were to be used in the generating plant.

The ventilation throughout was to be on the plenum system, the blowers overbalancing the exhausters with that end in view. The heating was to be on what is known as the direct-indirect system, and electric motors were to be used for all power purposes excepting for drip and return pumps, which were to exhaust directly into the heating stacks, and for the temperature-regulating system which used compressed air in common with the book and message despatch system for the library building, the exhaust steam from the power house to be used for heating the two large buildings on the major axis.

Direct current on the two-wire system was to be used for both power and light, with an initial potential of 120 volts.

The water distribution was to be on the same lines as that in use at the medical school, with an additional unit for fire protection.

When we began to get down to facts, as the buildings developed, it became apparent that the entire group would, when completed, require at least 4,000 nominal boiler horse-power and 1,500 electrical horse-power for their operation on a normal basis, requiring a coal consumption of 15,000 tons per annum.

The service furnished by the power house covered a period of eight months, when the university is in full operation, and a period of four months of vacation, when only the library building is in use, and the demand for light and power would be

reduced to a minimum. The varying conditions of weather, as well as the intermittent use of the buildings during term time on Saturdays and holidays, called for a wide range in boiler power and engine units, in order to attain economical operation.

To the end that there need be no necessity for subsequent

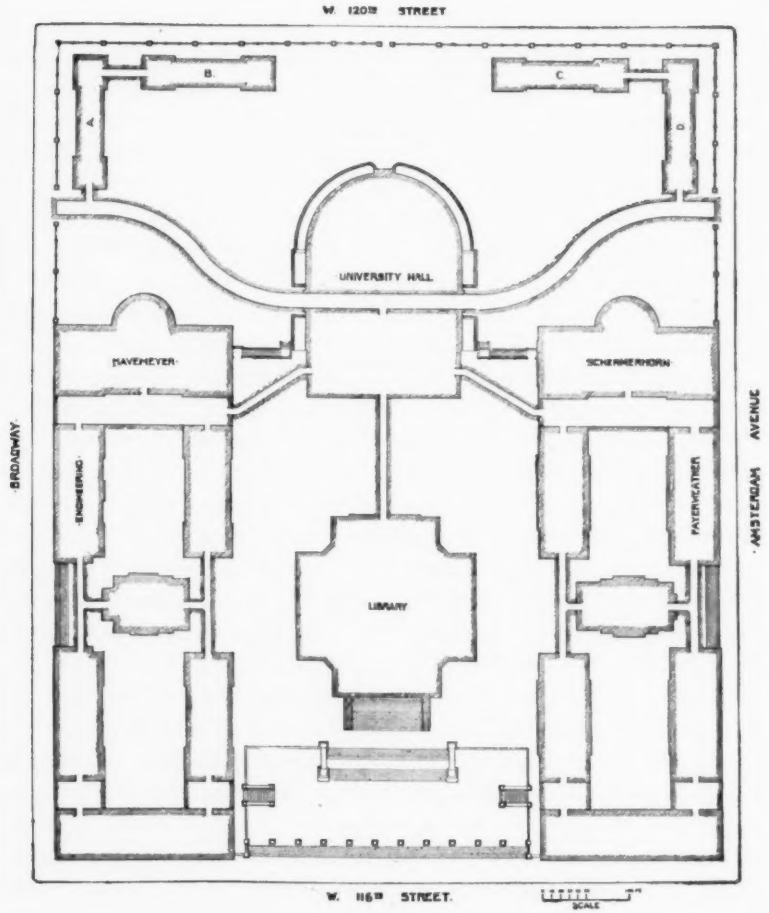
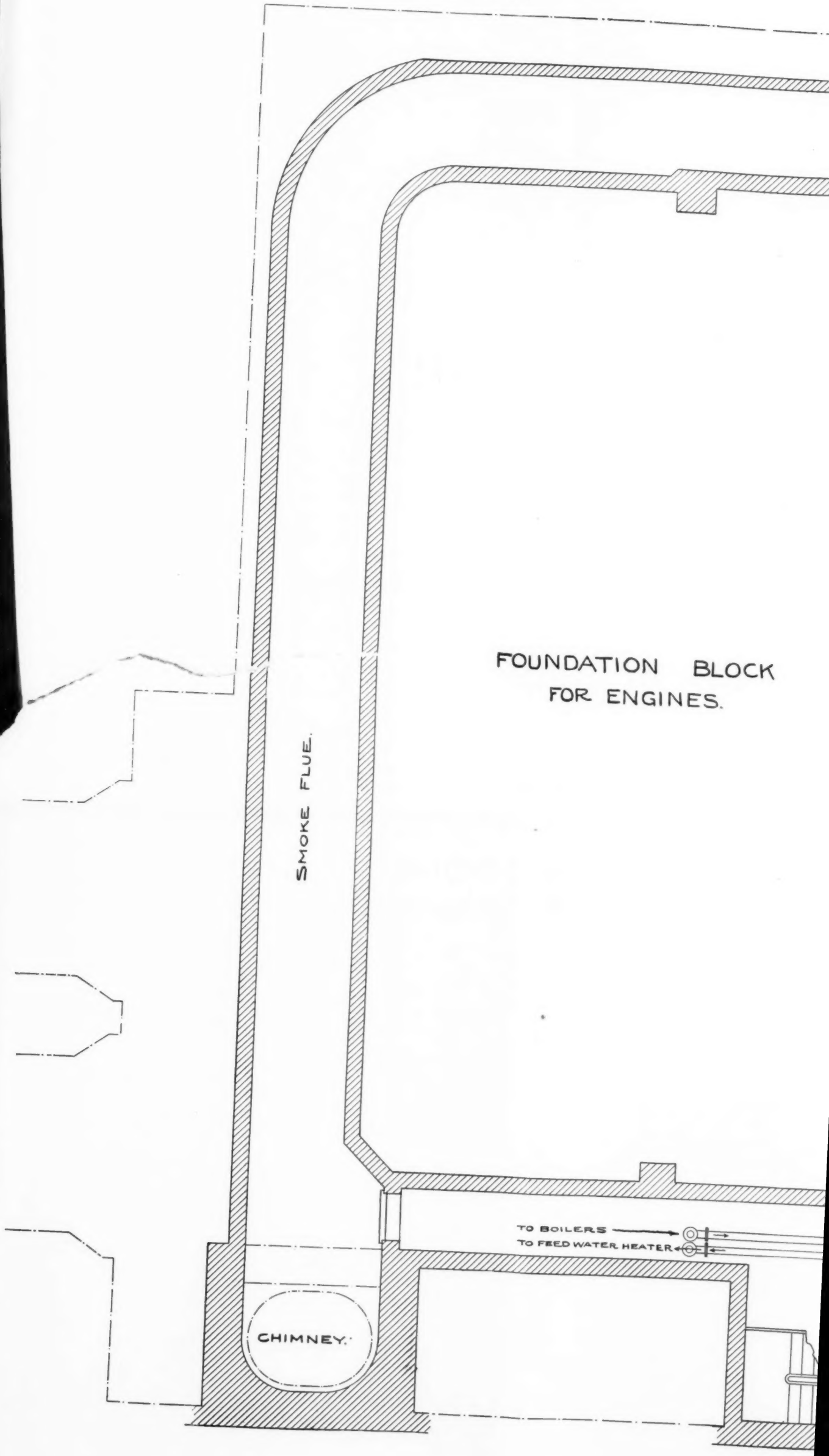
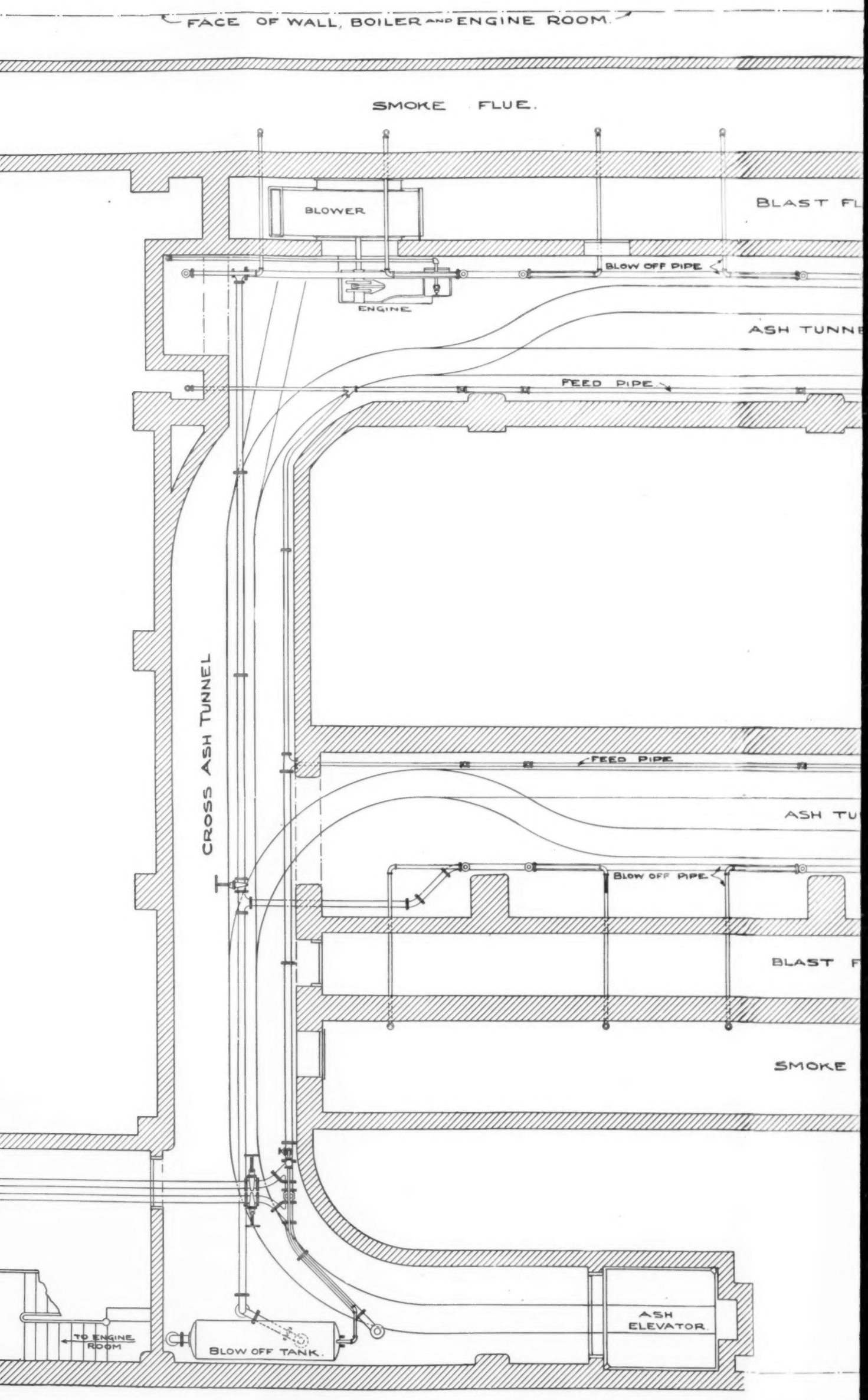


FIG. 241.

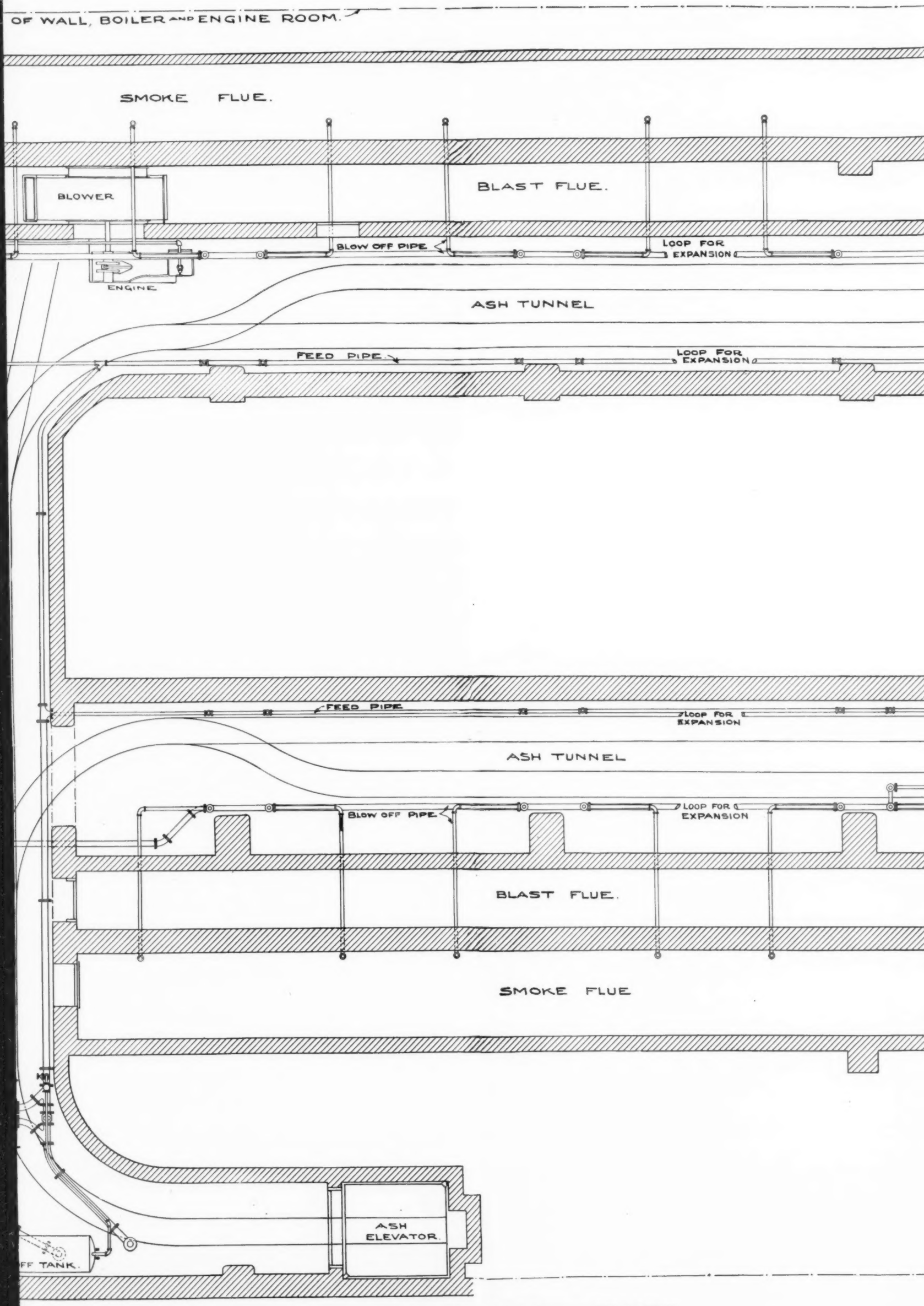
cutting or alteration of the building, the writer decided to lay down the complete plant in detail, using the best types of standard commercial apparatus, to build the emplacements for the entire plant, and put in at once so much of it as would be needed to operate the seven buildings then in course of erection


FOUNDATION BLOCK
FOR ENGINES.
SMOKE FLUE.
TO BOILERS
TO FEED WATER HEATER
CHIMNEY.

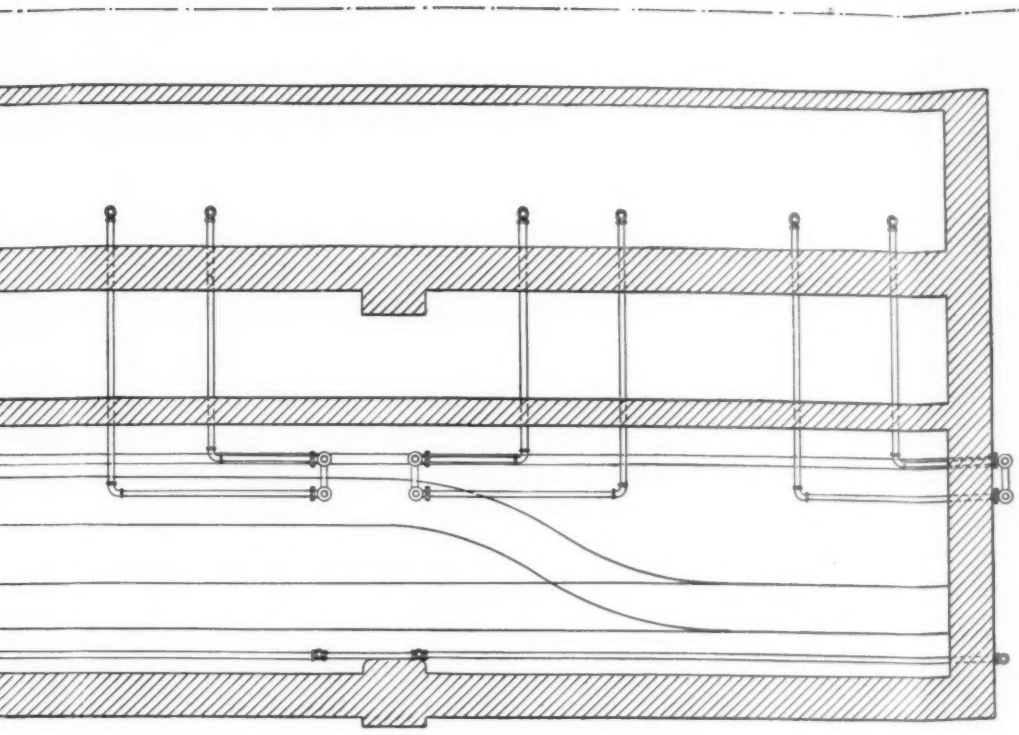


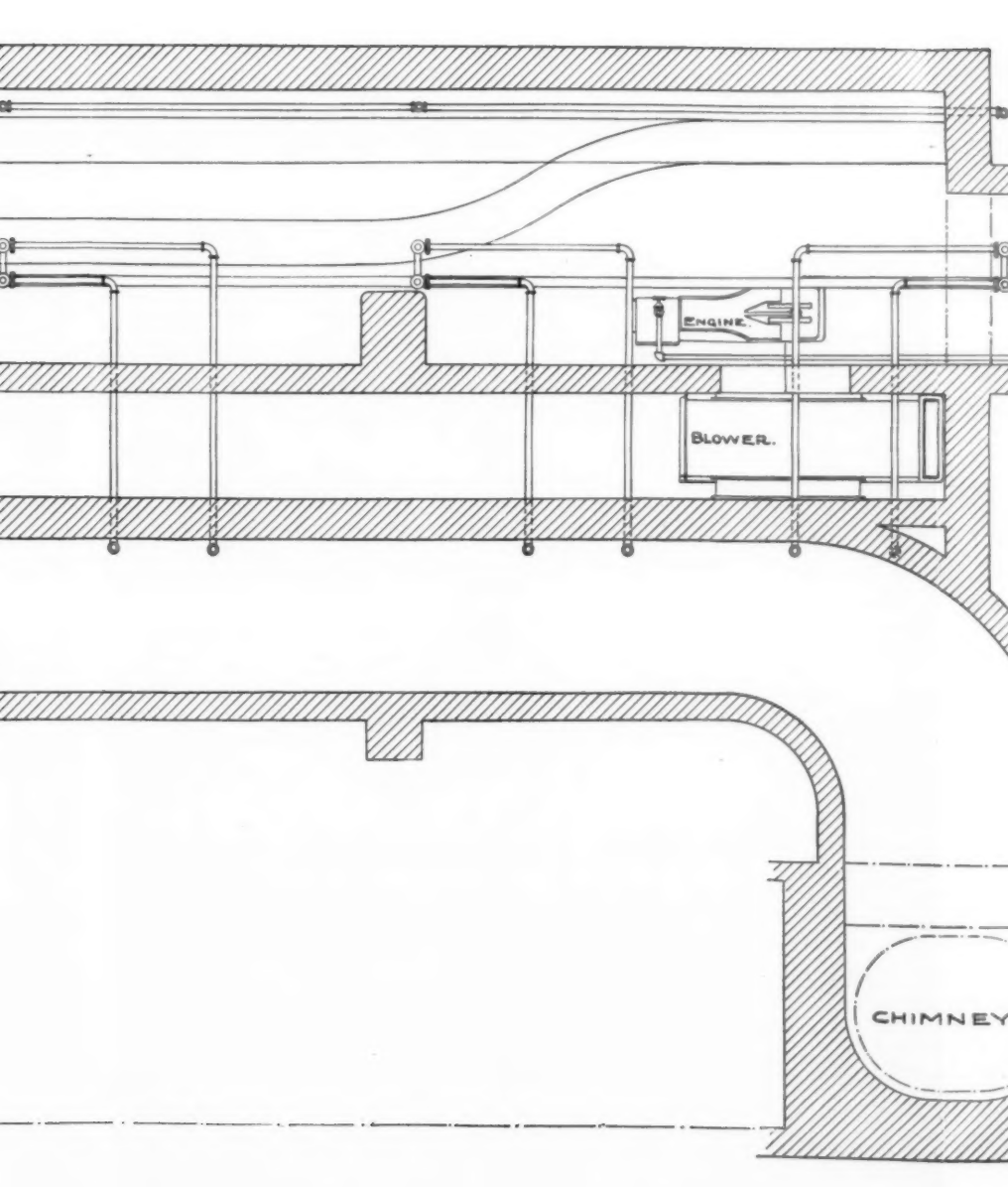
PLAN OF ASH TUNNELS AN

FIG. 242.



PLAN OF ASH TUNNELS AND FLUES





HEAT, LIGHT AND POWER PLANT
COLUMBIA UNIVERSITY
116TH STREET SITE.

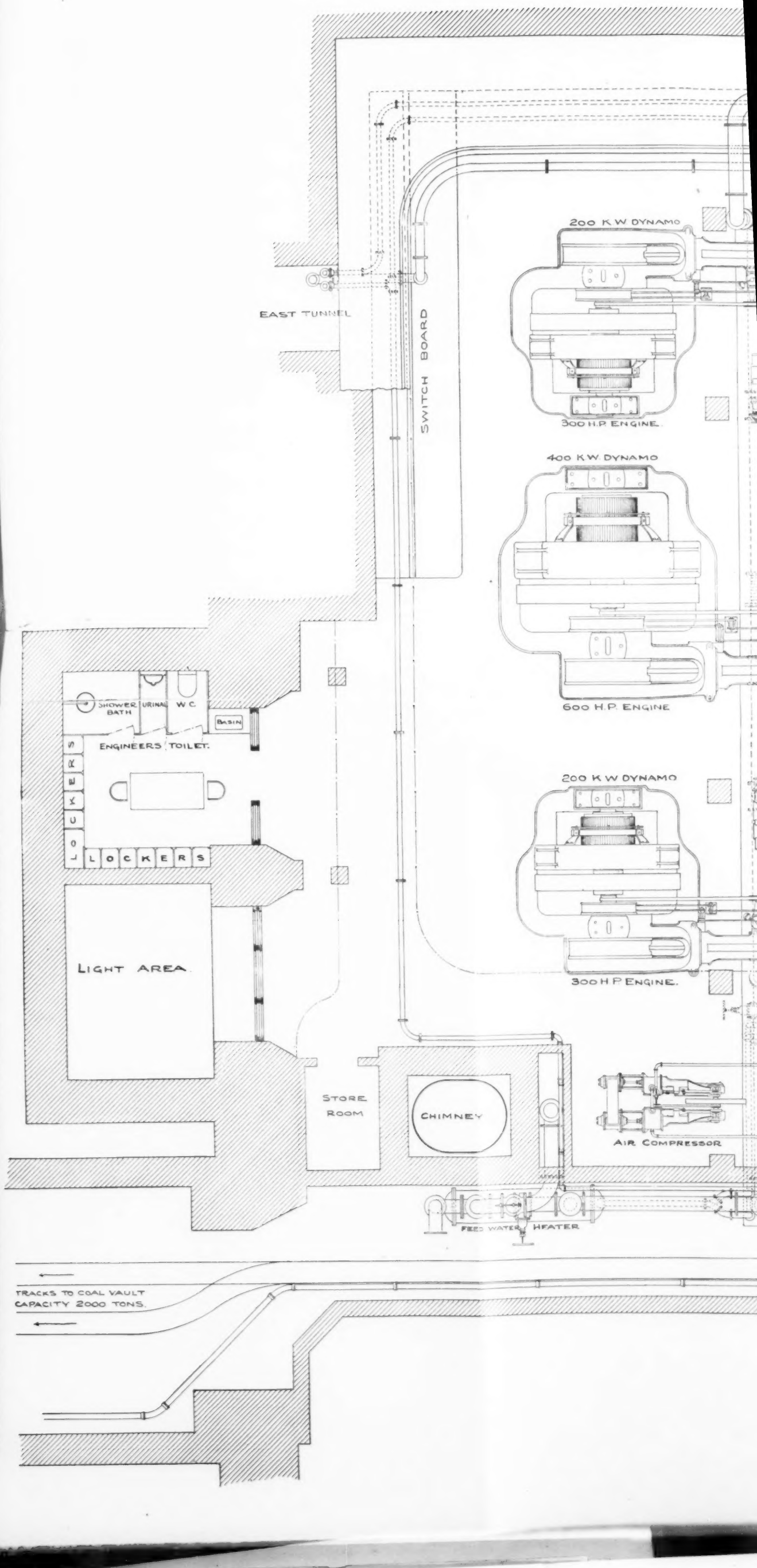
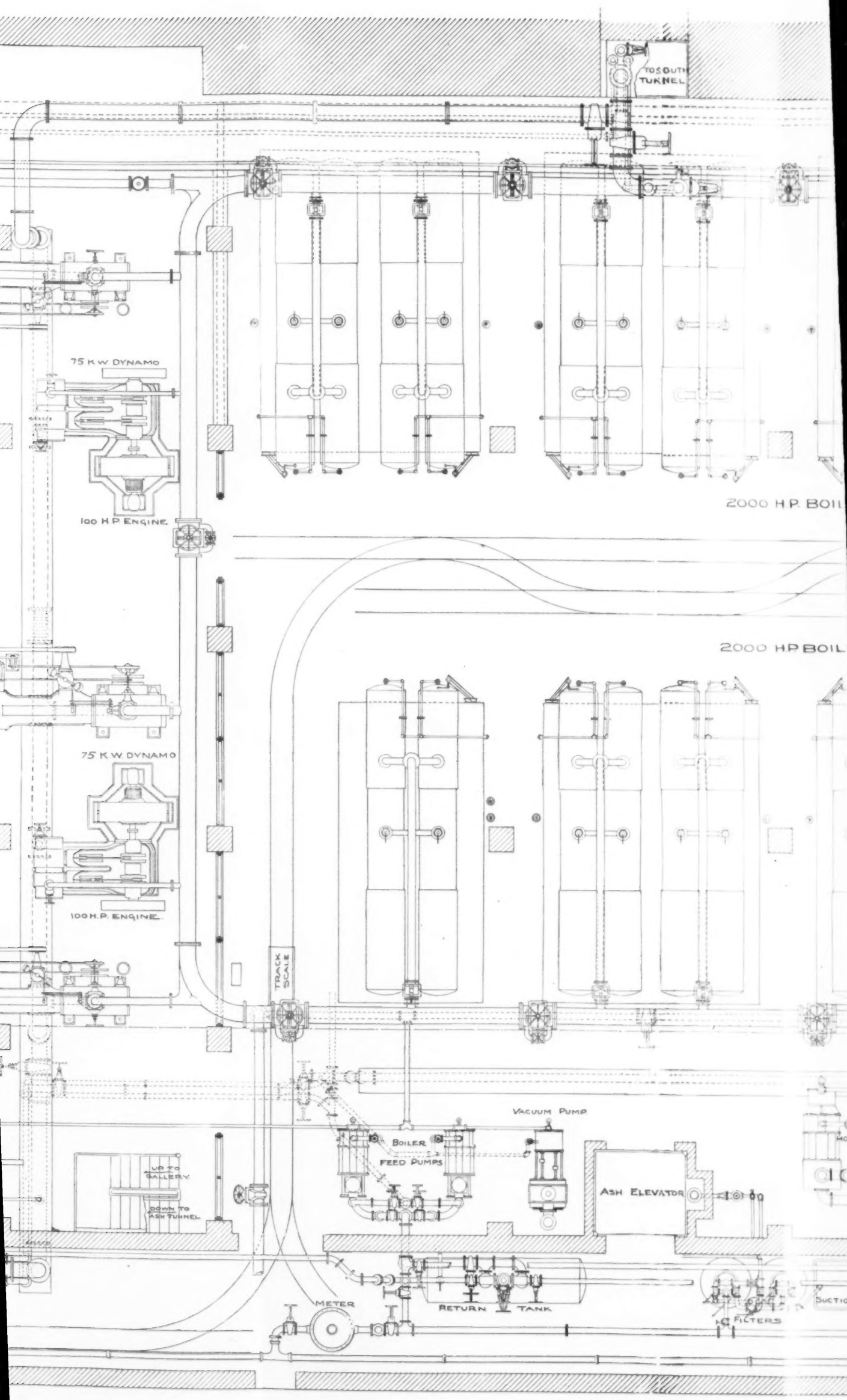
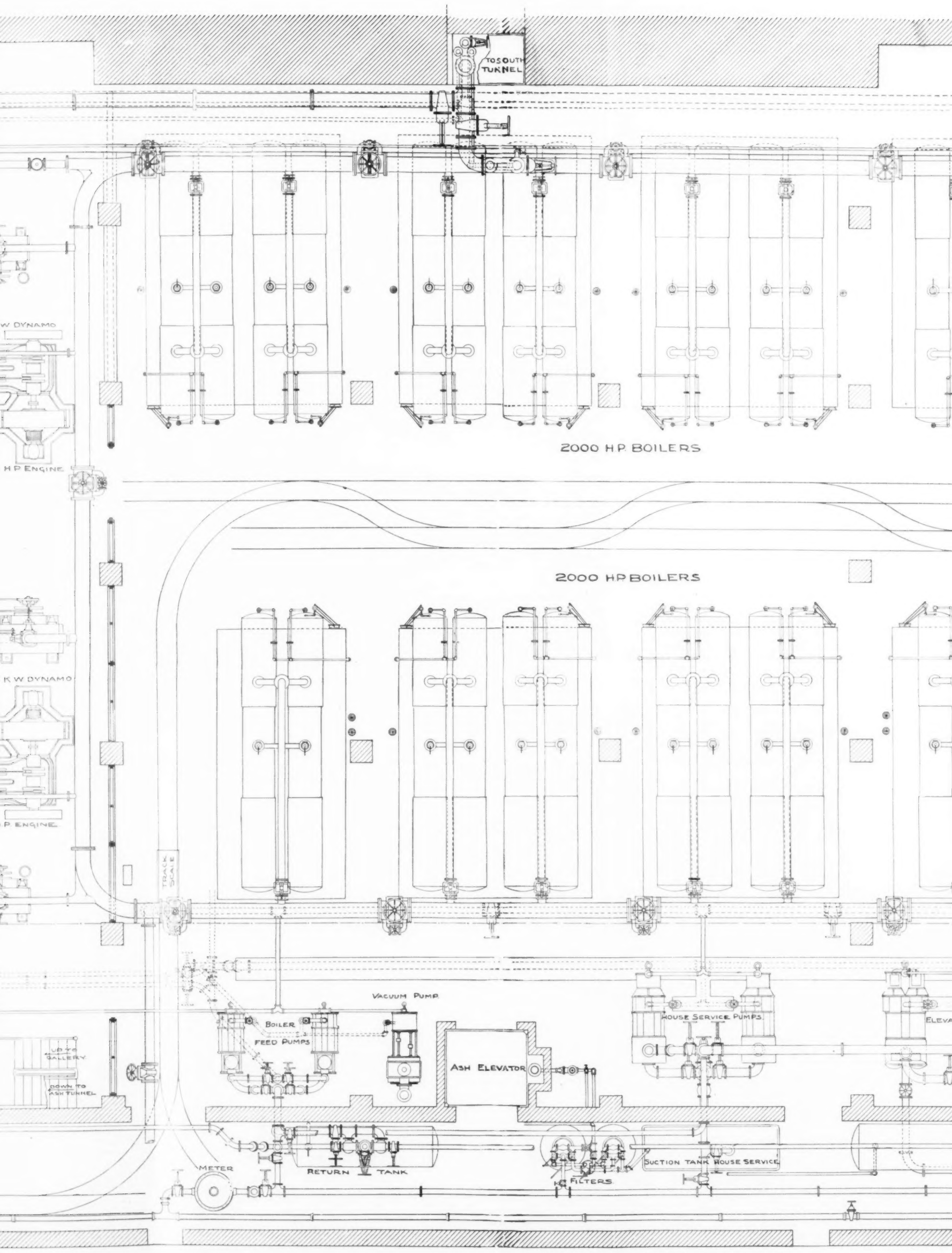

200 K.W. DYNAMO
EAST TUNNEL
SWITCH BOARD
300 H.P. ENGINE.
400 K.W. DYNAMO
600 H.P. ENGINE
SHOWER BATH
URINAL
W.C.
BASIN
ENGINEERS TOILET.
LOCKERS
LOCKERS
200 K.W. DYNAMO
300 H.P. ENGINE.
LIGHT AREA.
STORE ROOM
CHIMNEY
AIR COMPRESSOR
FEED WATER HEATER
TRACKS TO COAL VAULT
CAPACITY 2000 TONS.

Fig. 243.

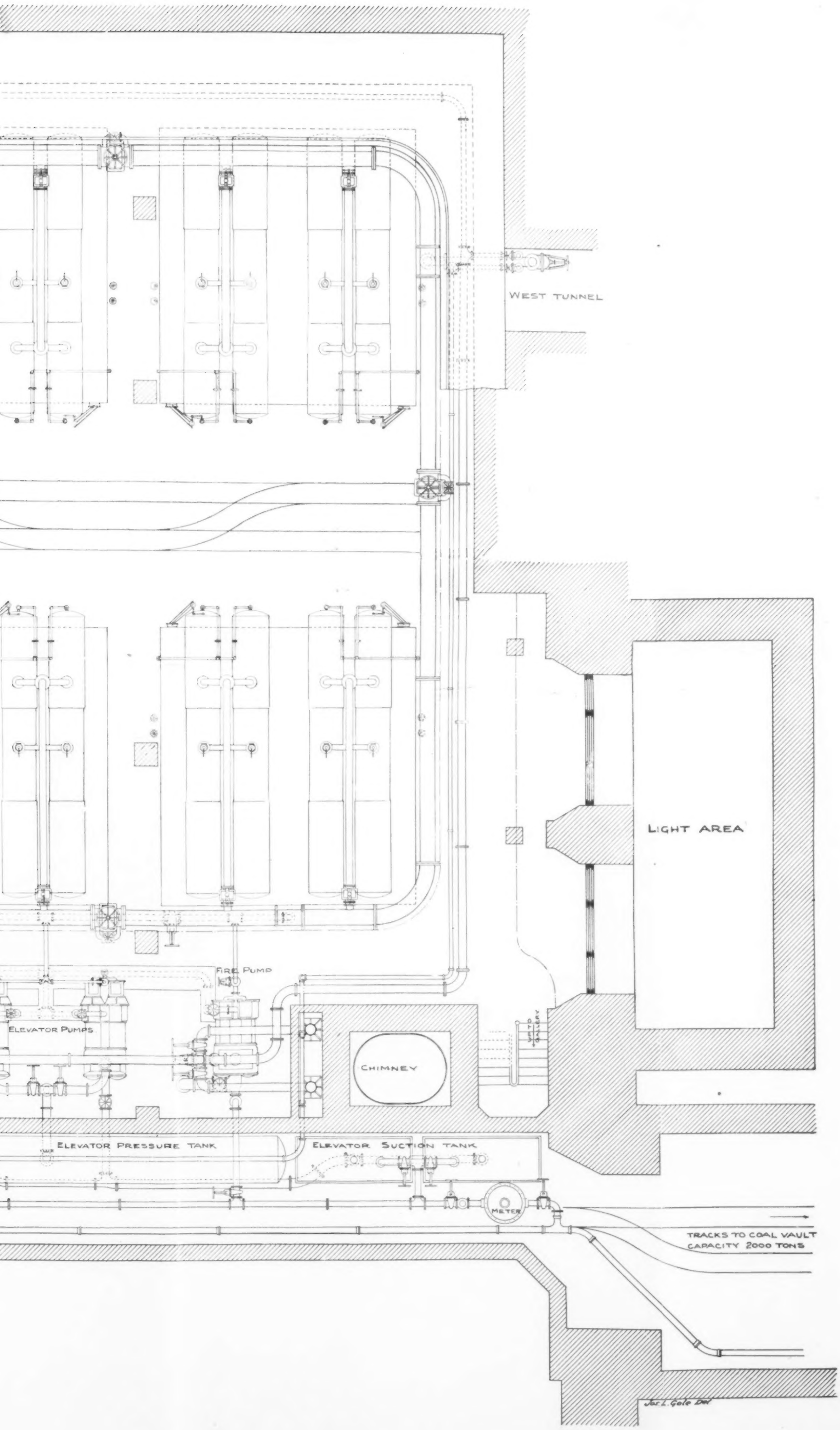


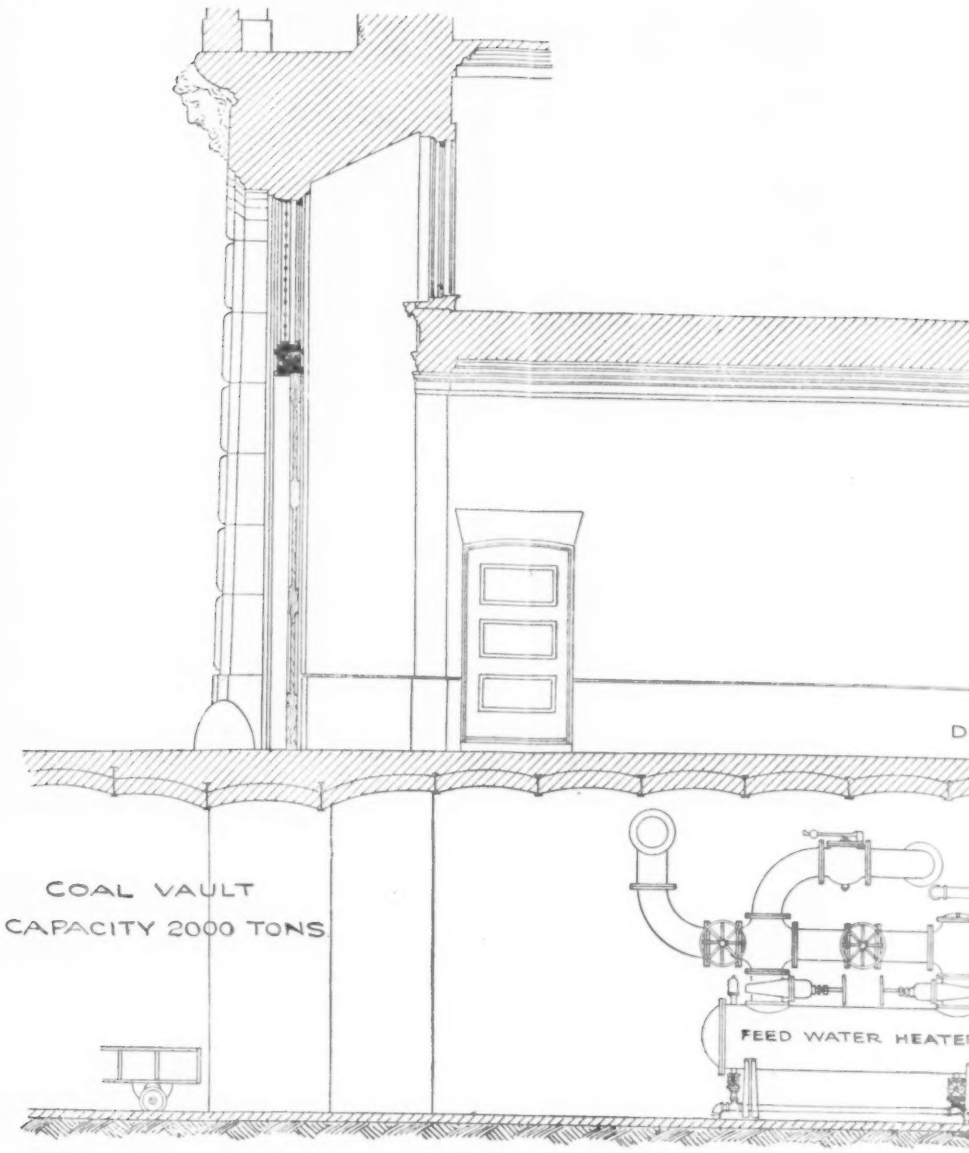
PLAN OF ENGINE AND BOILER ROOMS

Fig. 243.

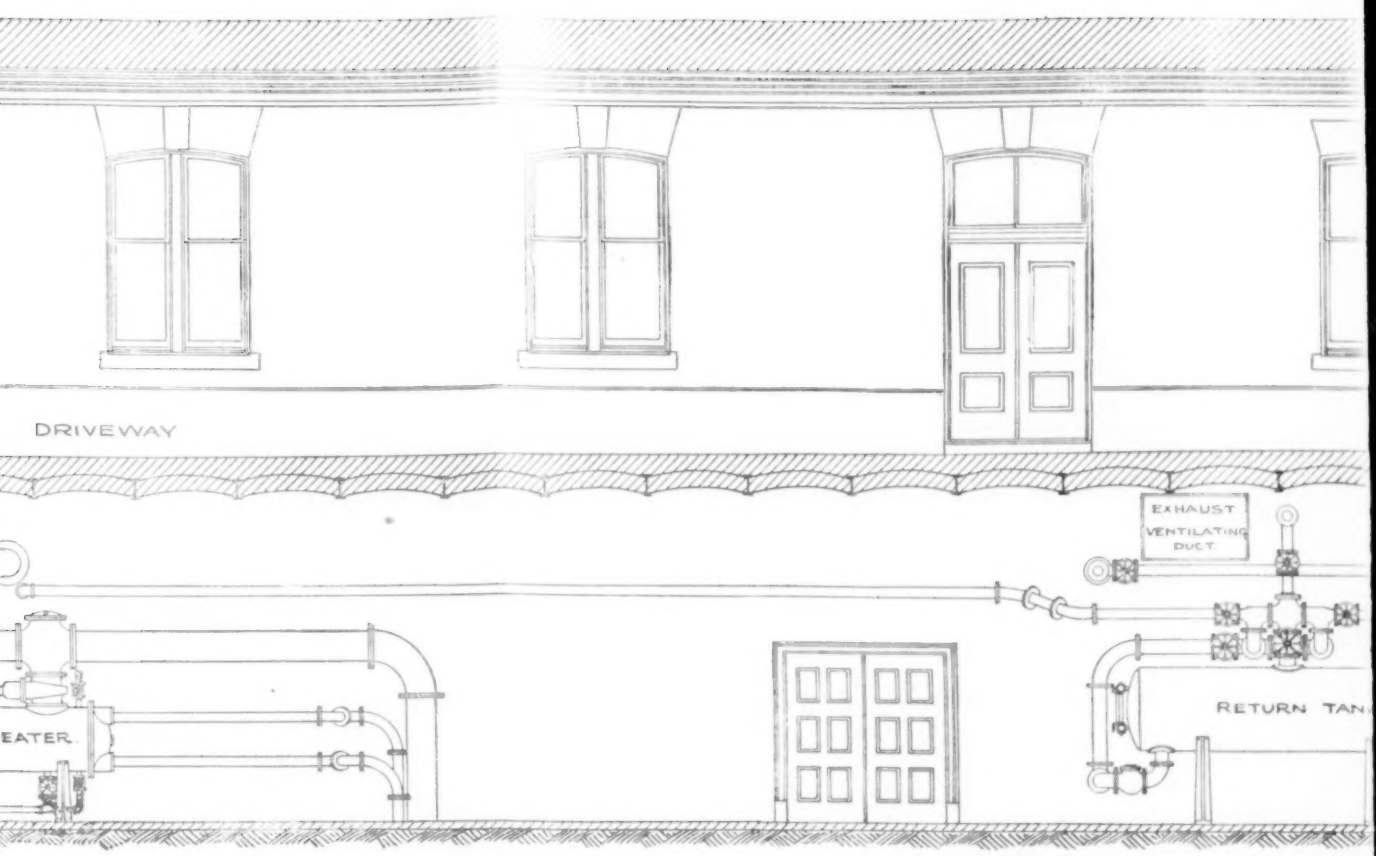


PLAN OF ENGINE AND BOILER ROOMS

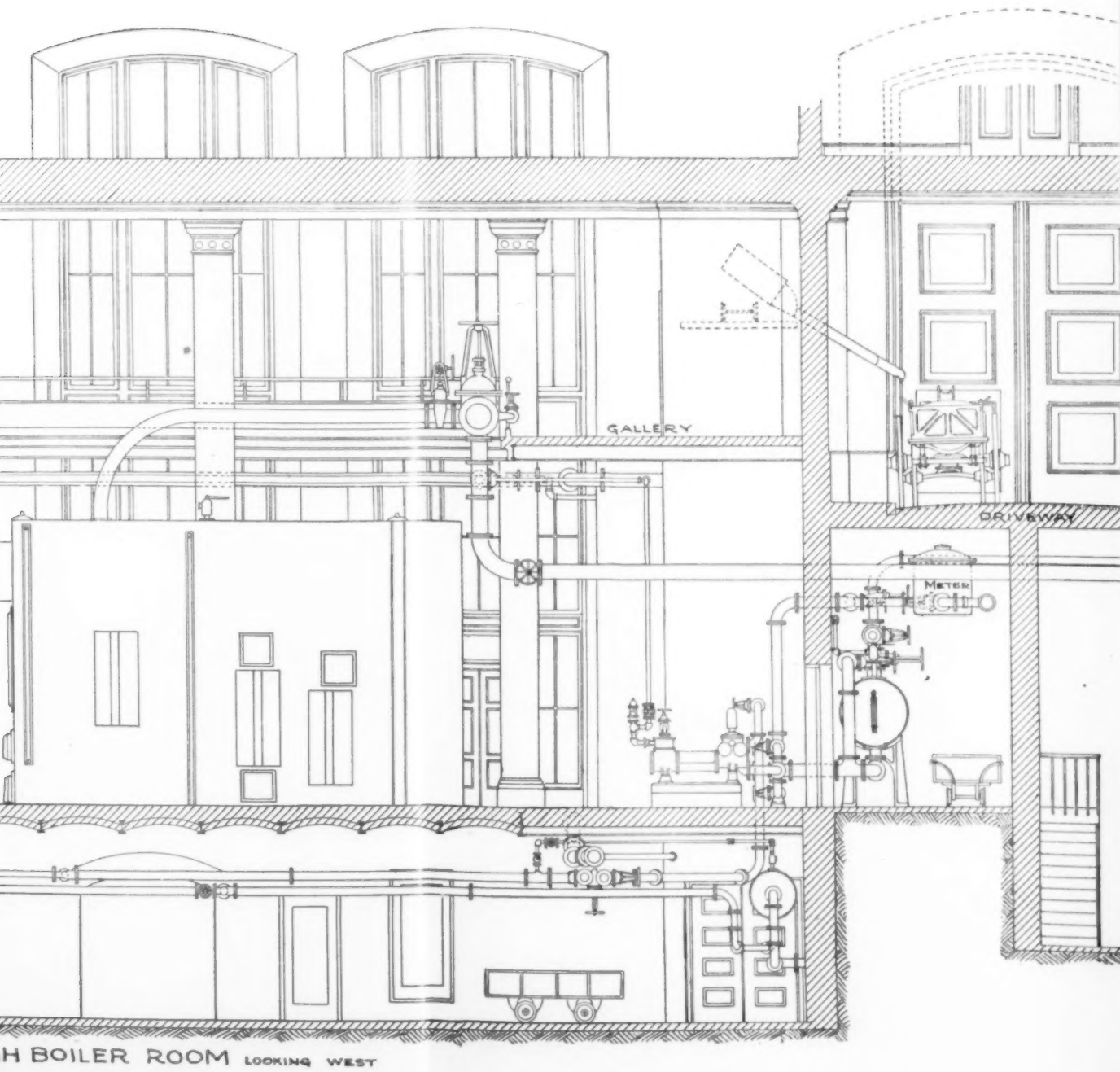

WEST TUNNEL
LIGHT AREA
FIRE PUMP
ELEVATOR PUMPS
CHIMNEY
ELEVATOR PRESSURE TANK
ELEVATOR SUCTION TANK
METER
TRACKS TO COAL VAULT
CAPACITY 2000 TONS
Jos L. Gale Del



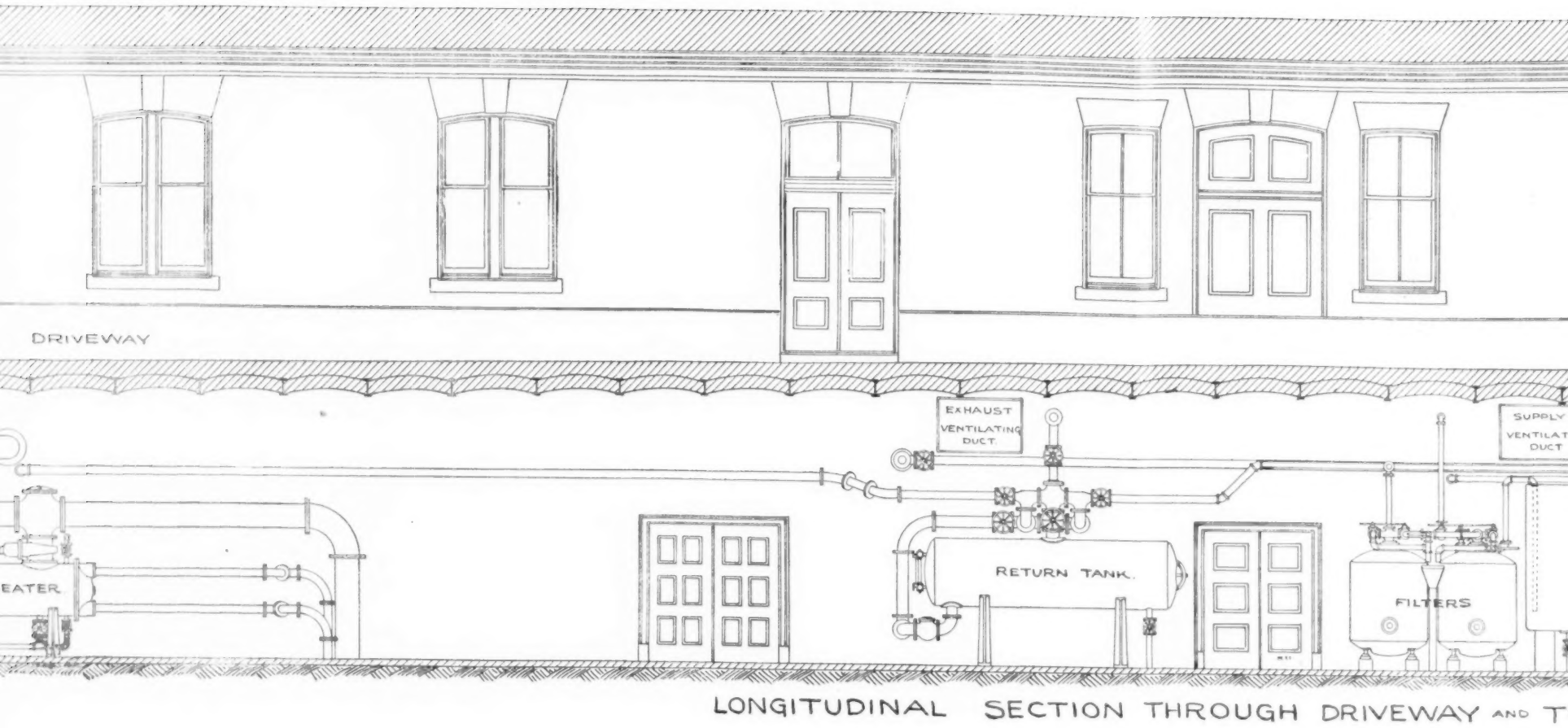
TRANSVERSE SECTION THROUGH BOI



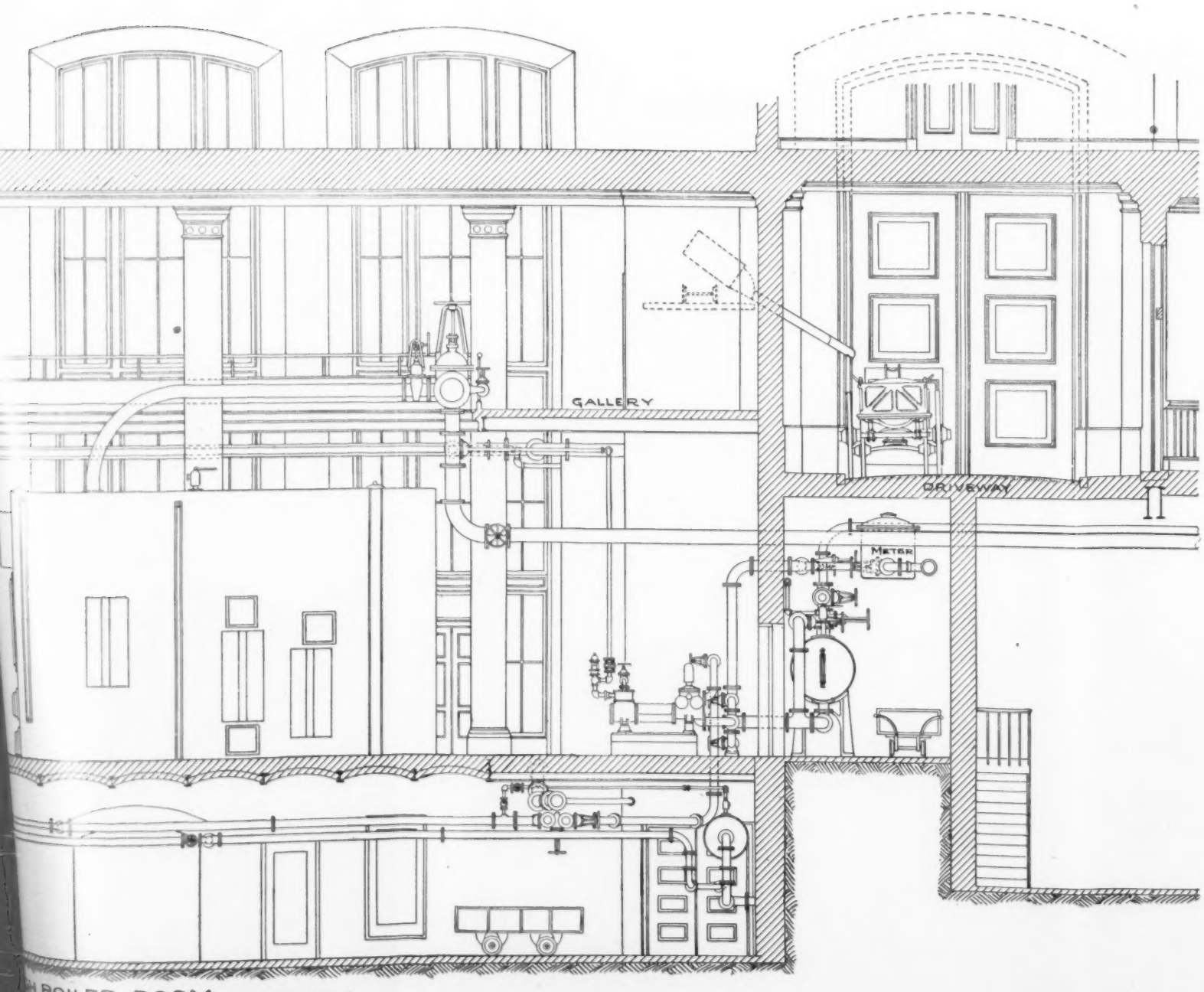
LONGITUDINAL SECTION



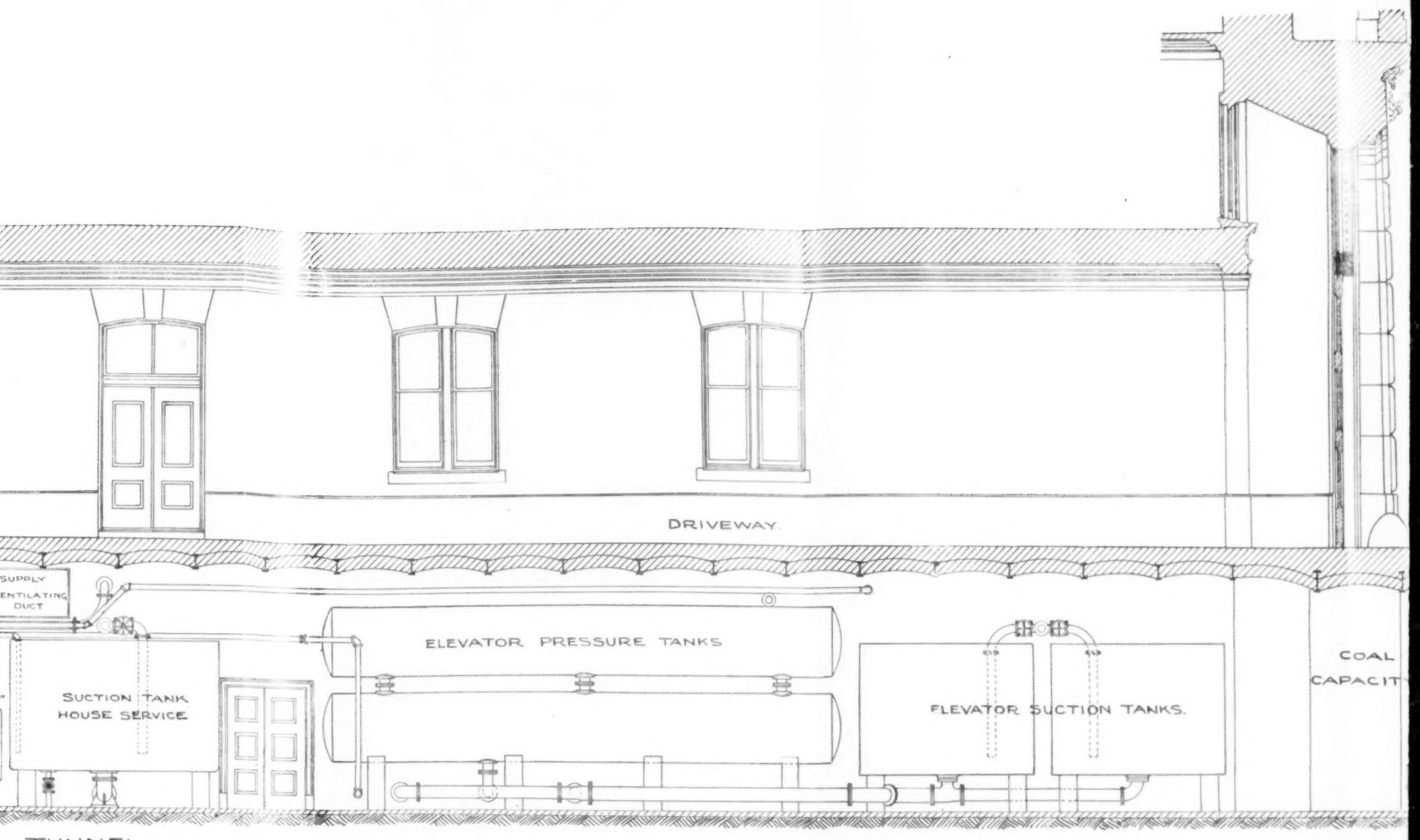
H BOILER ROOM LOOKING WEST



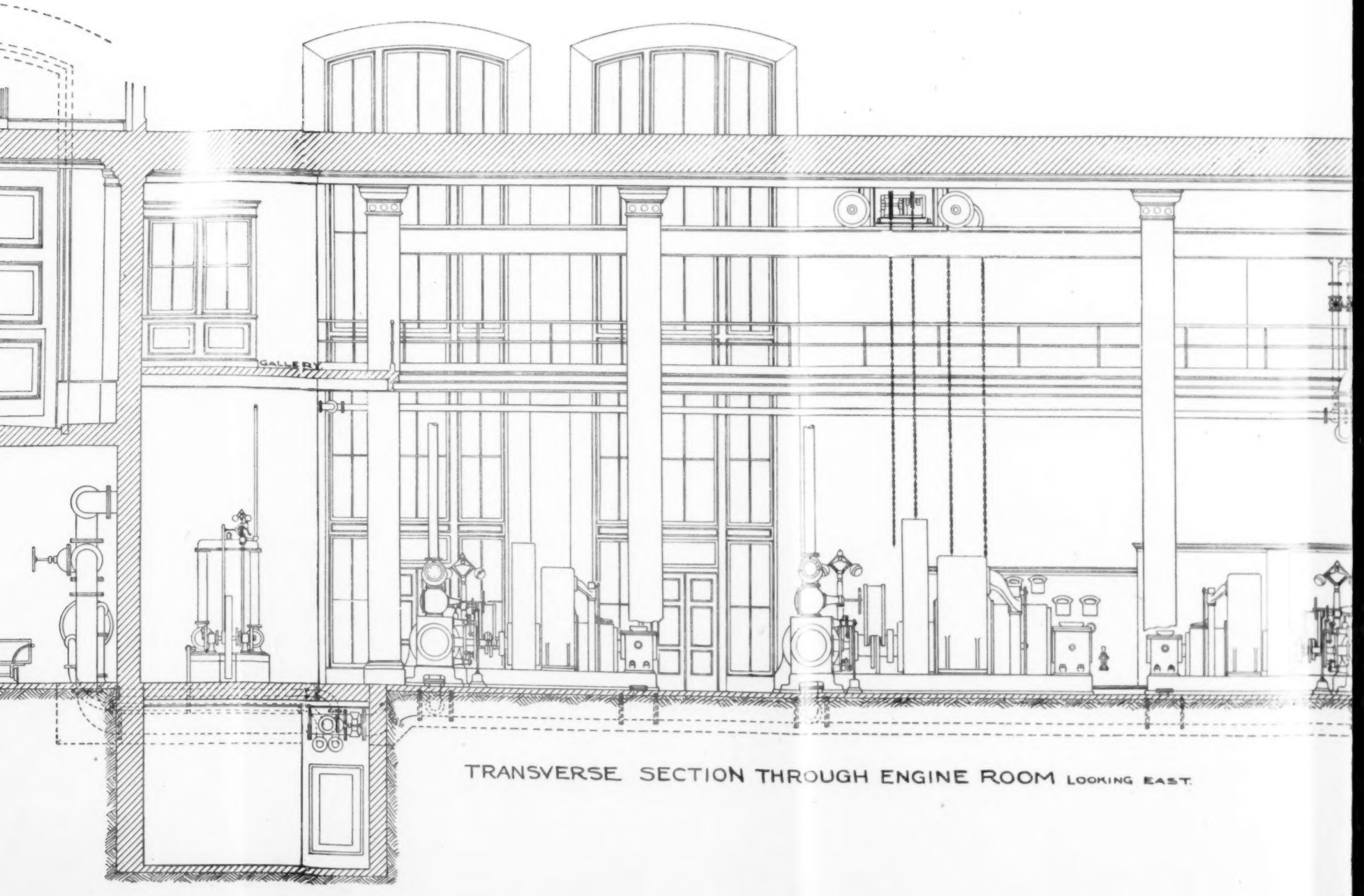
LONGITUDINAL SECTION THROUGH DRIVEWAY AND T



H BOILER ROOM LOOKING WEST

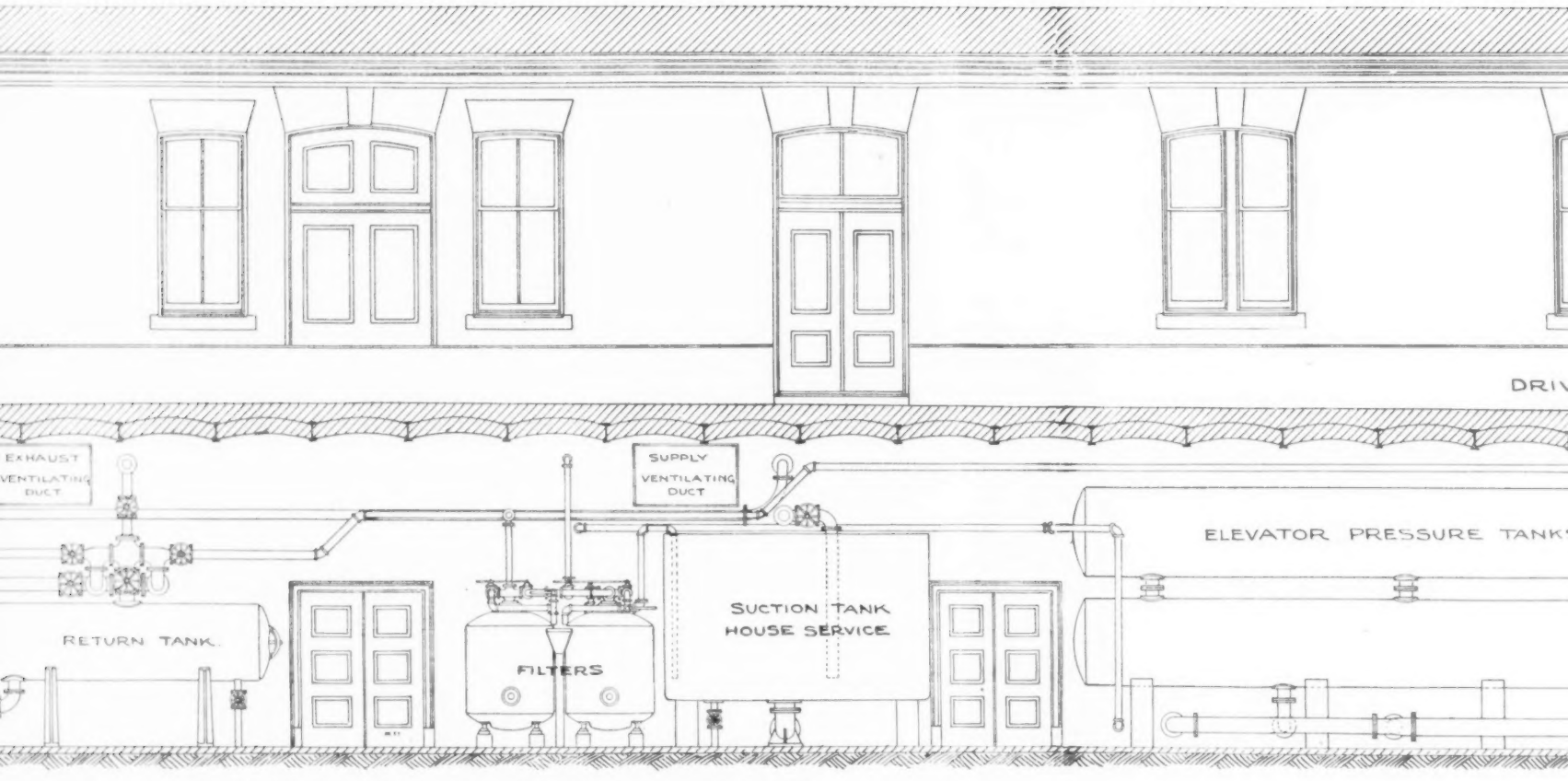


D TUNNEL LOOKING SOUTH

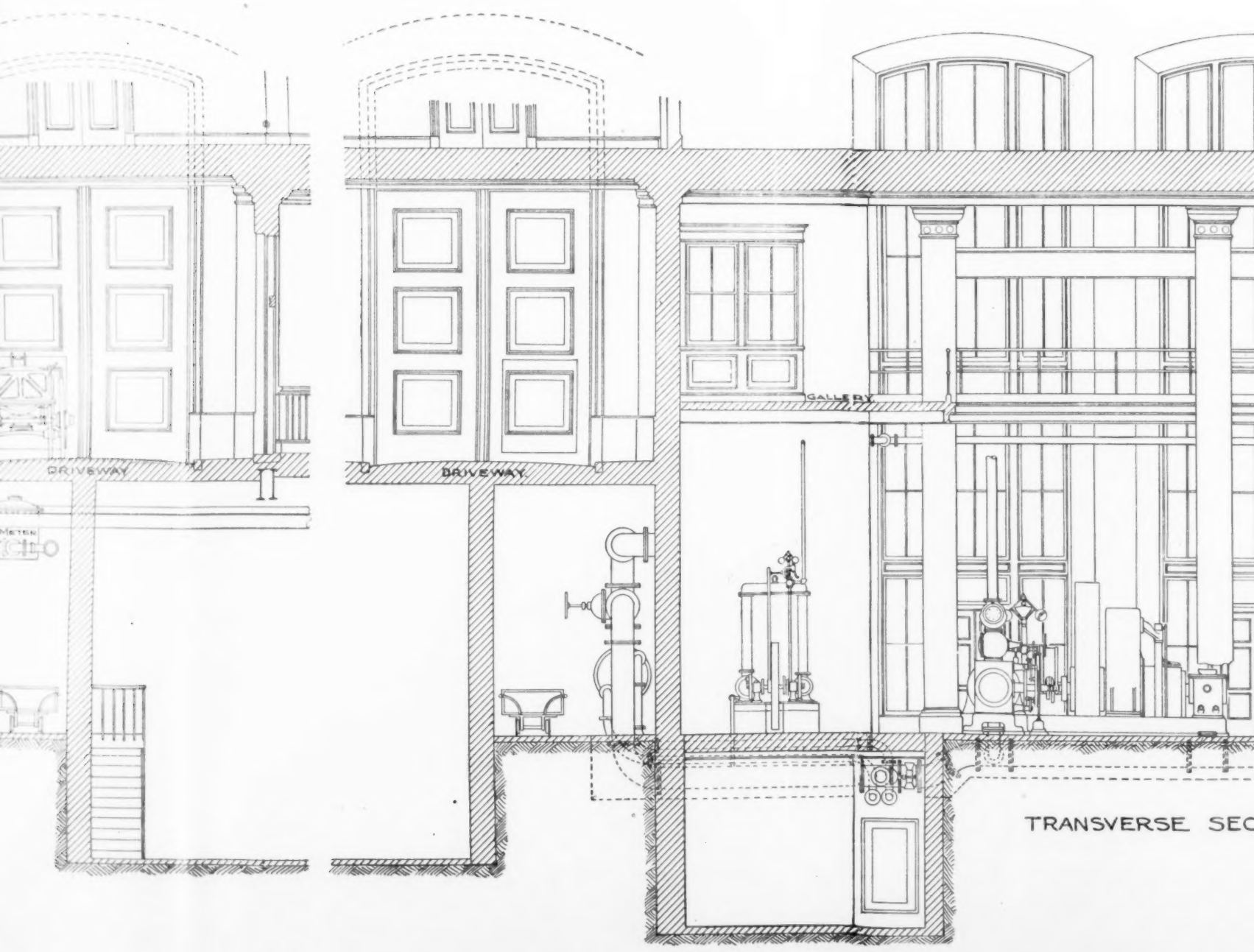


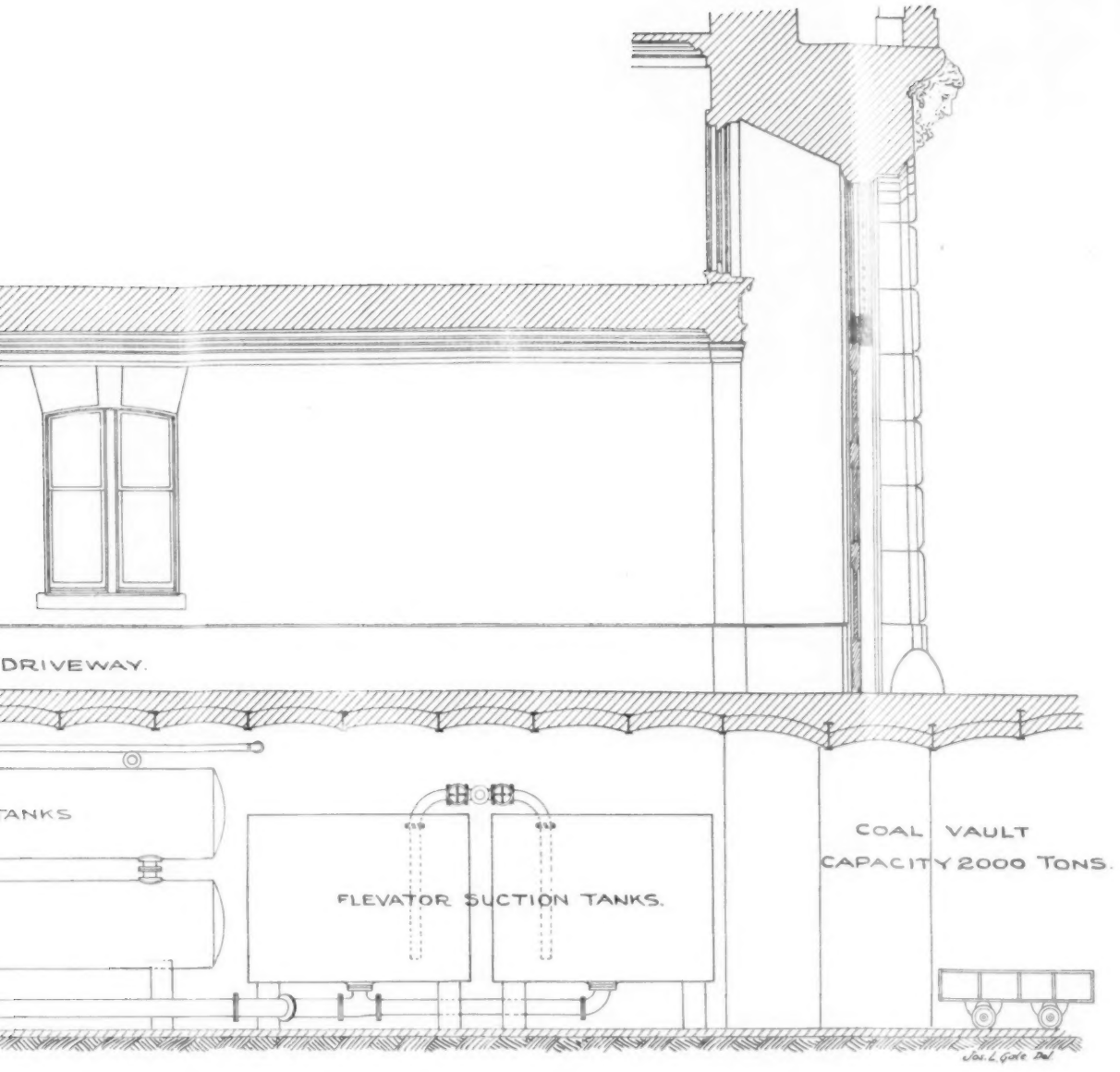
TRANSVERSE SECTION THROUGH ENGINE ROOM LOOKING EAST.

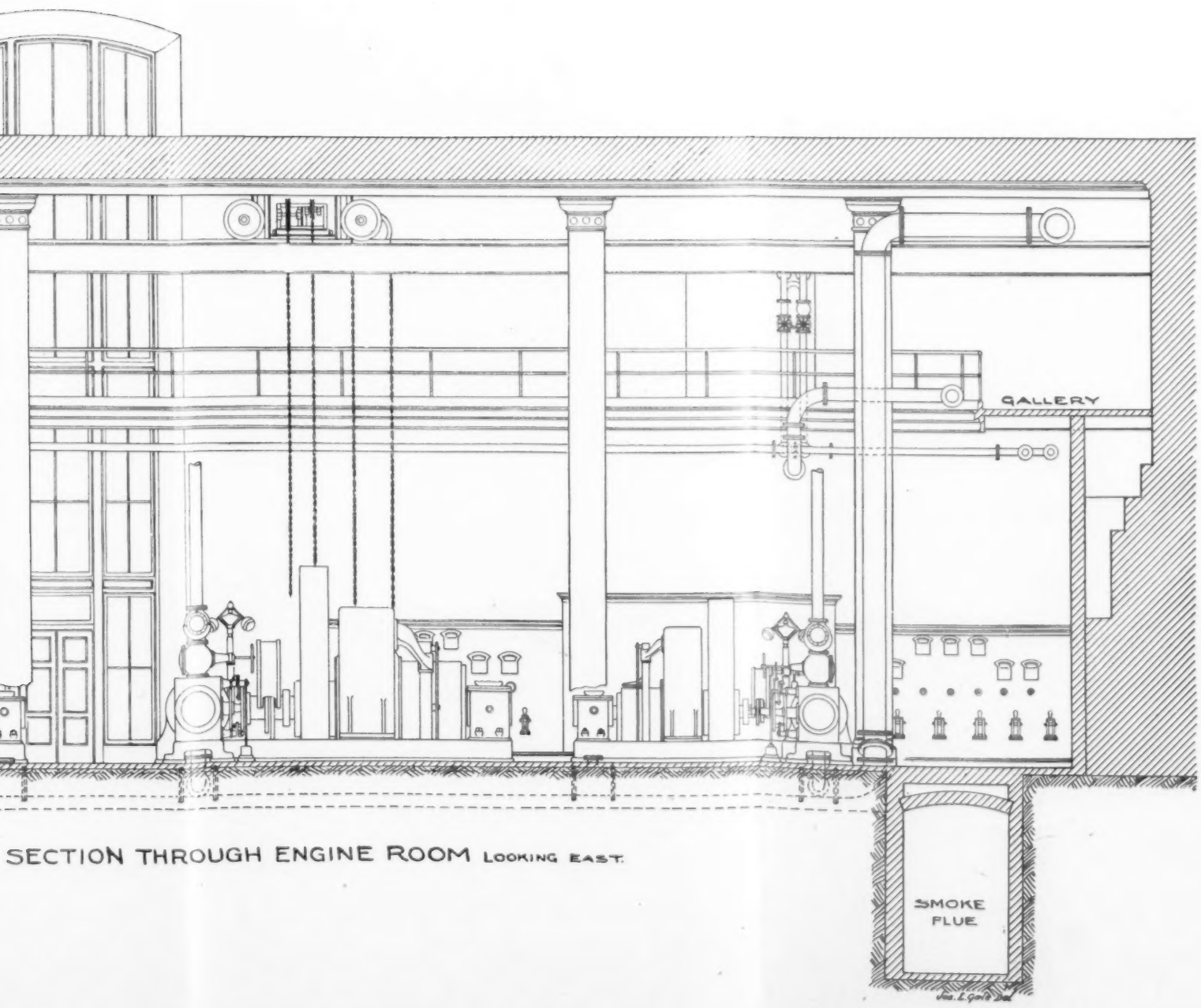
Fig. 244.



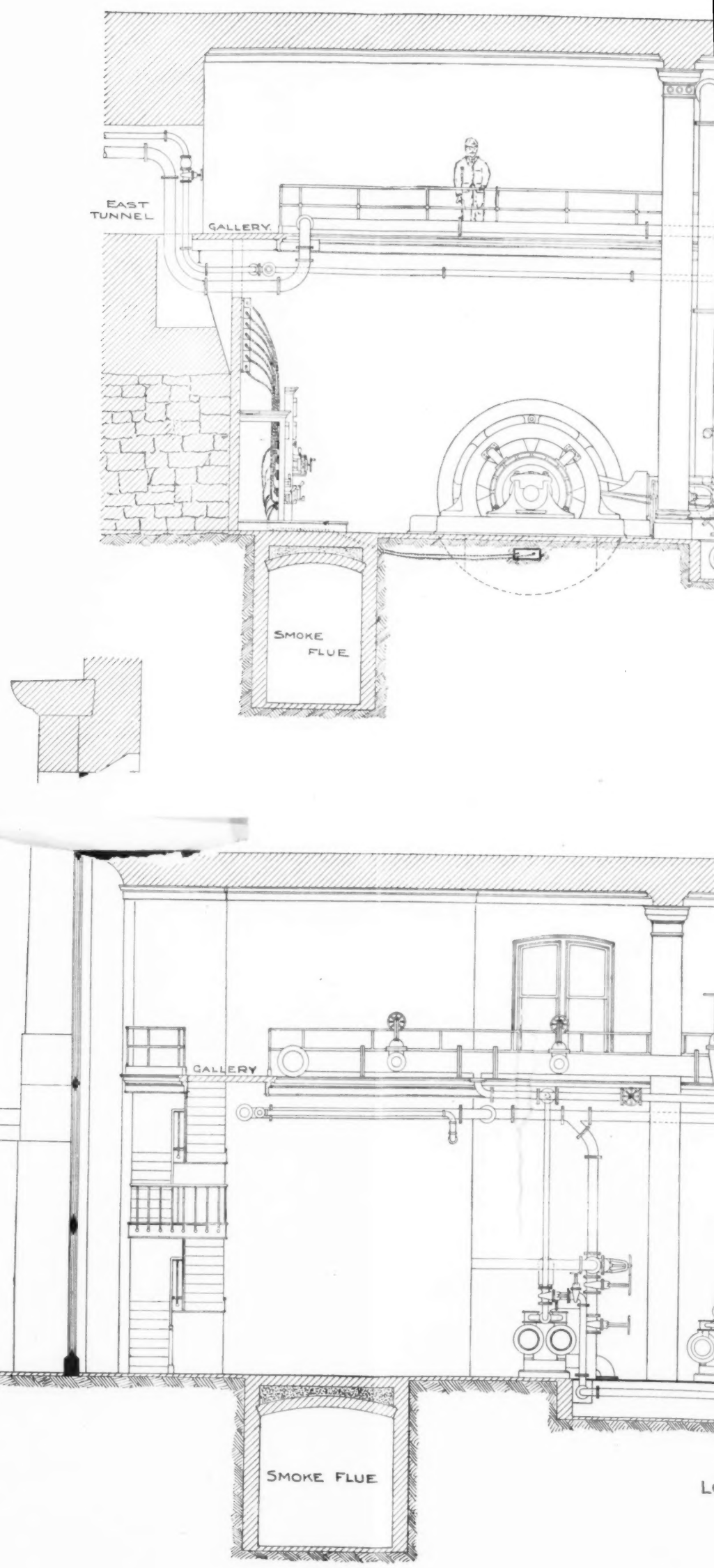
SECTION THROUGH DRIVEWAY AND TUNNEL LOOKING SOUTH

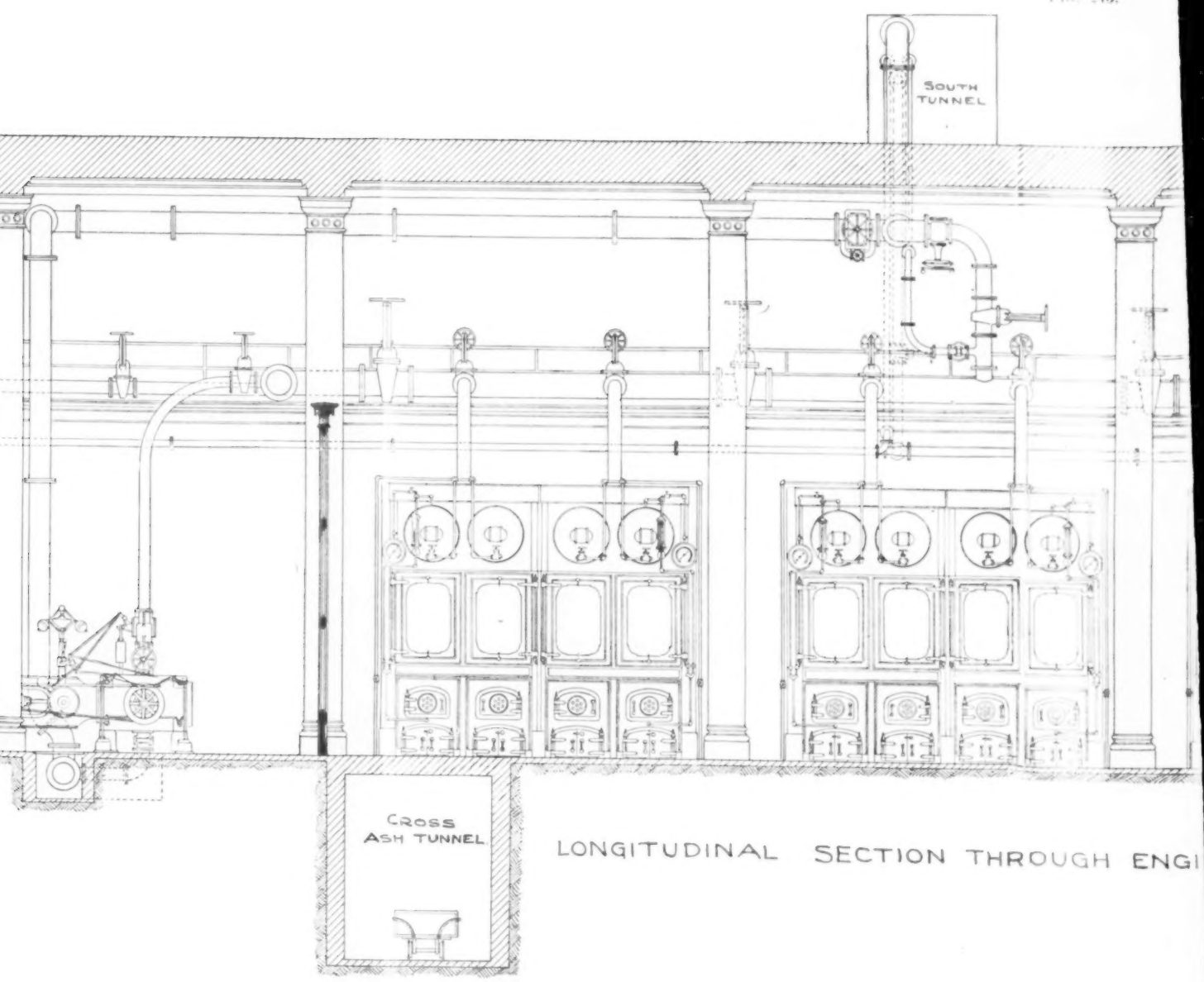




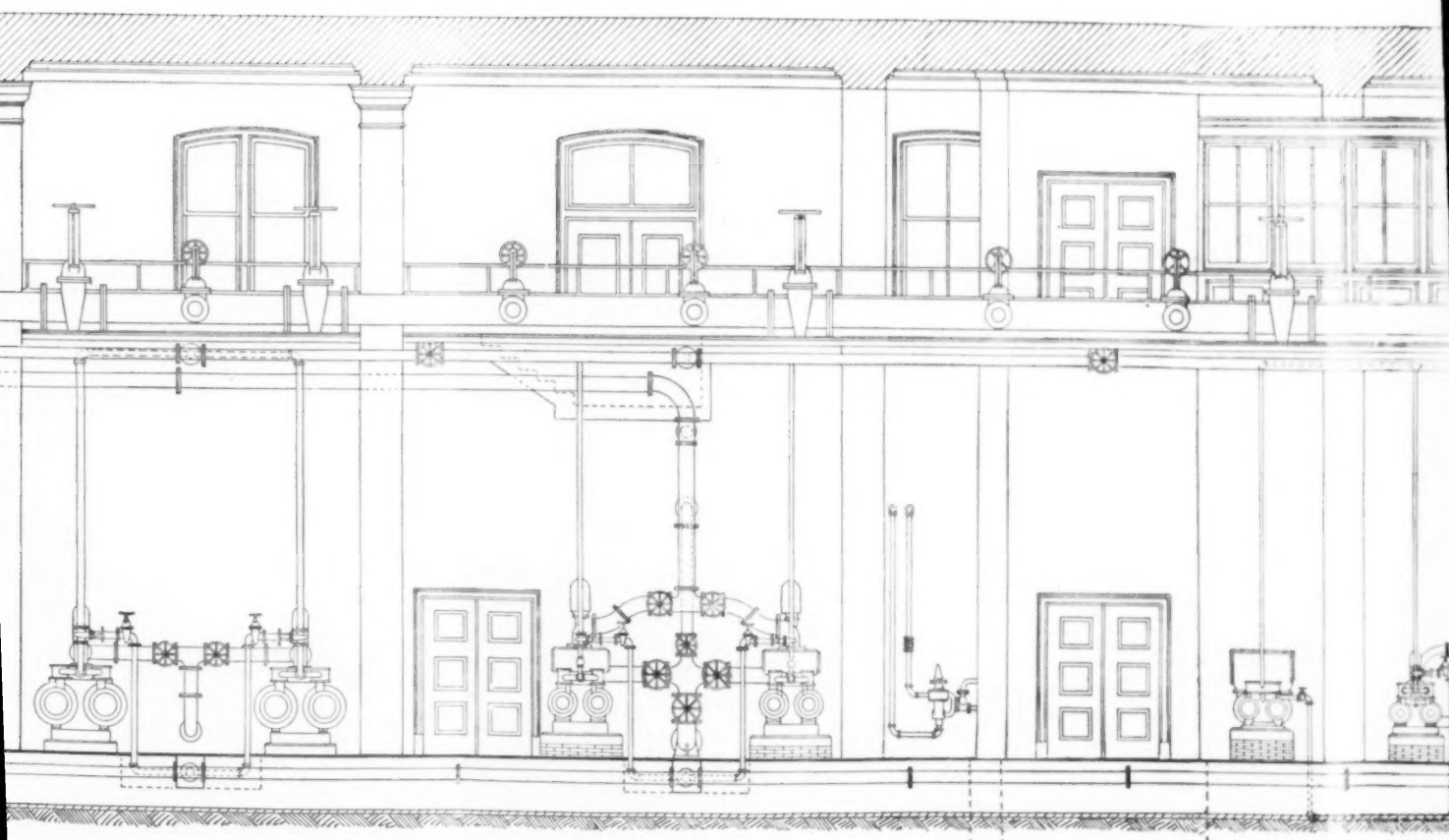


SECTION THROUGH ENGINE ROOM LOOKING EAST.


EAST
TUNNEL
GALLERY.
SMOKE
FLUE
GALLERY
LIGHT AREA
SMOKE FLUE

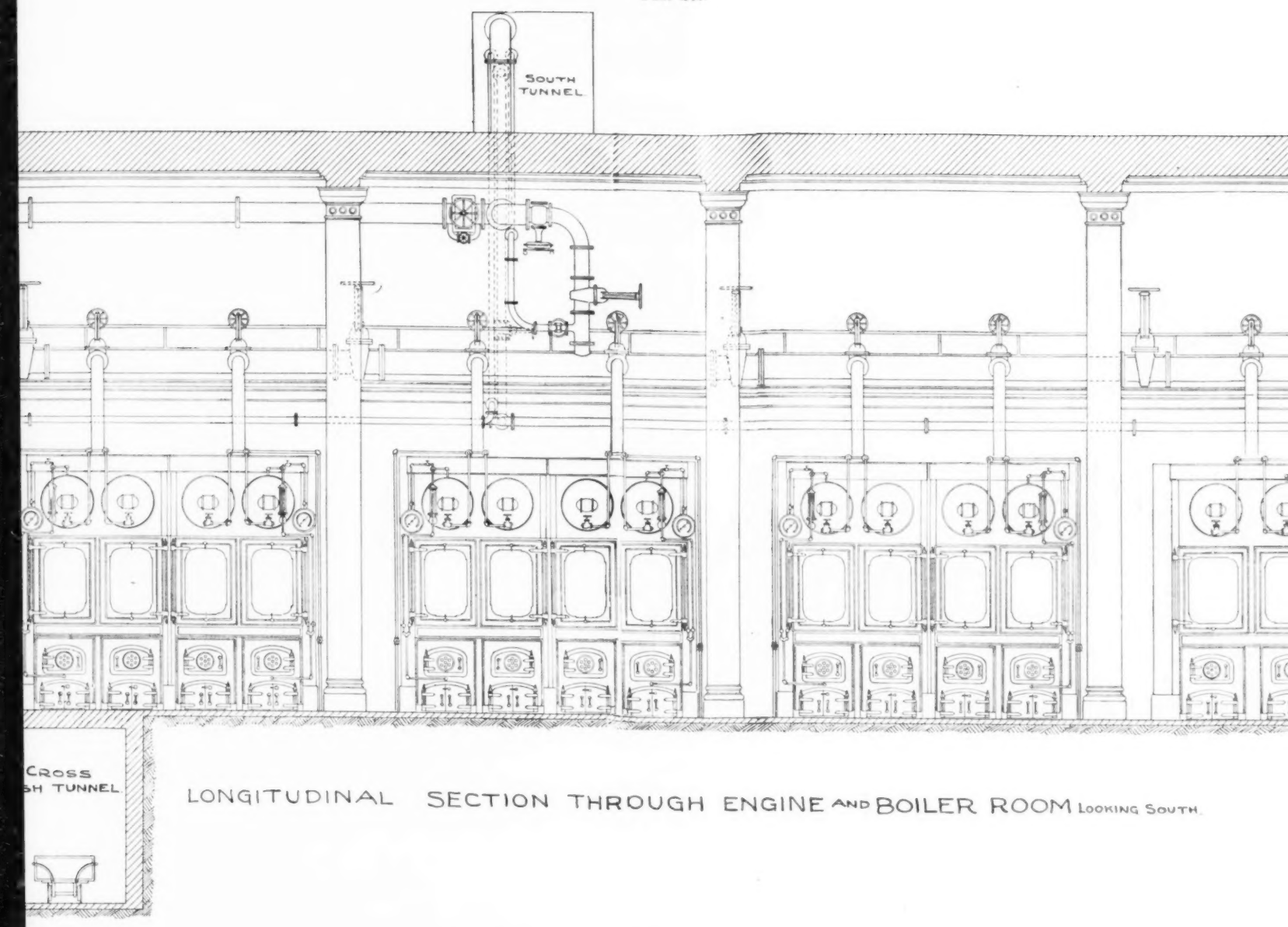


LONGITUDINAL SECTION THROUGH ENGI

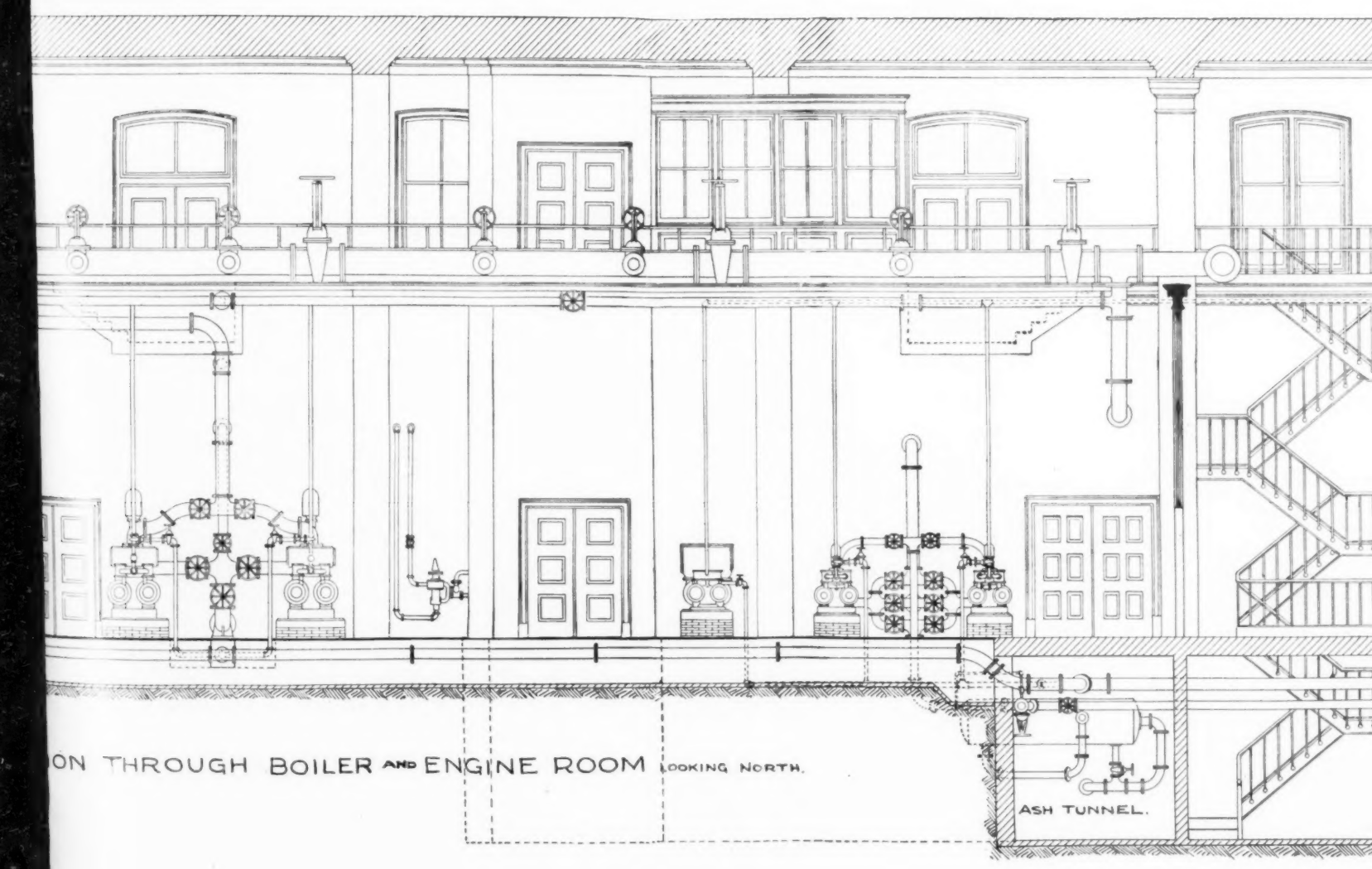


LONGITUDINAL SECTION THROUGH BOILER AND ENGINE ROOM LOOKING NORTH.

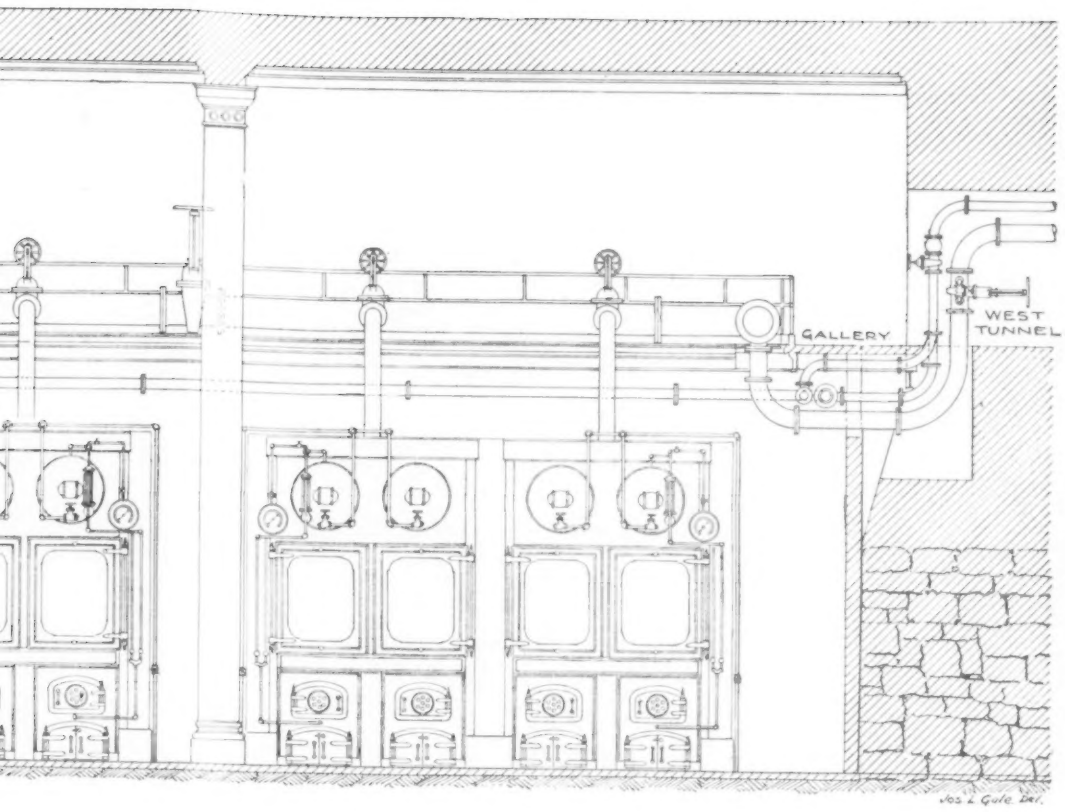
FIG. 245

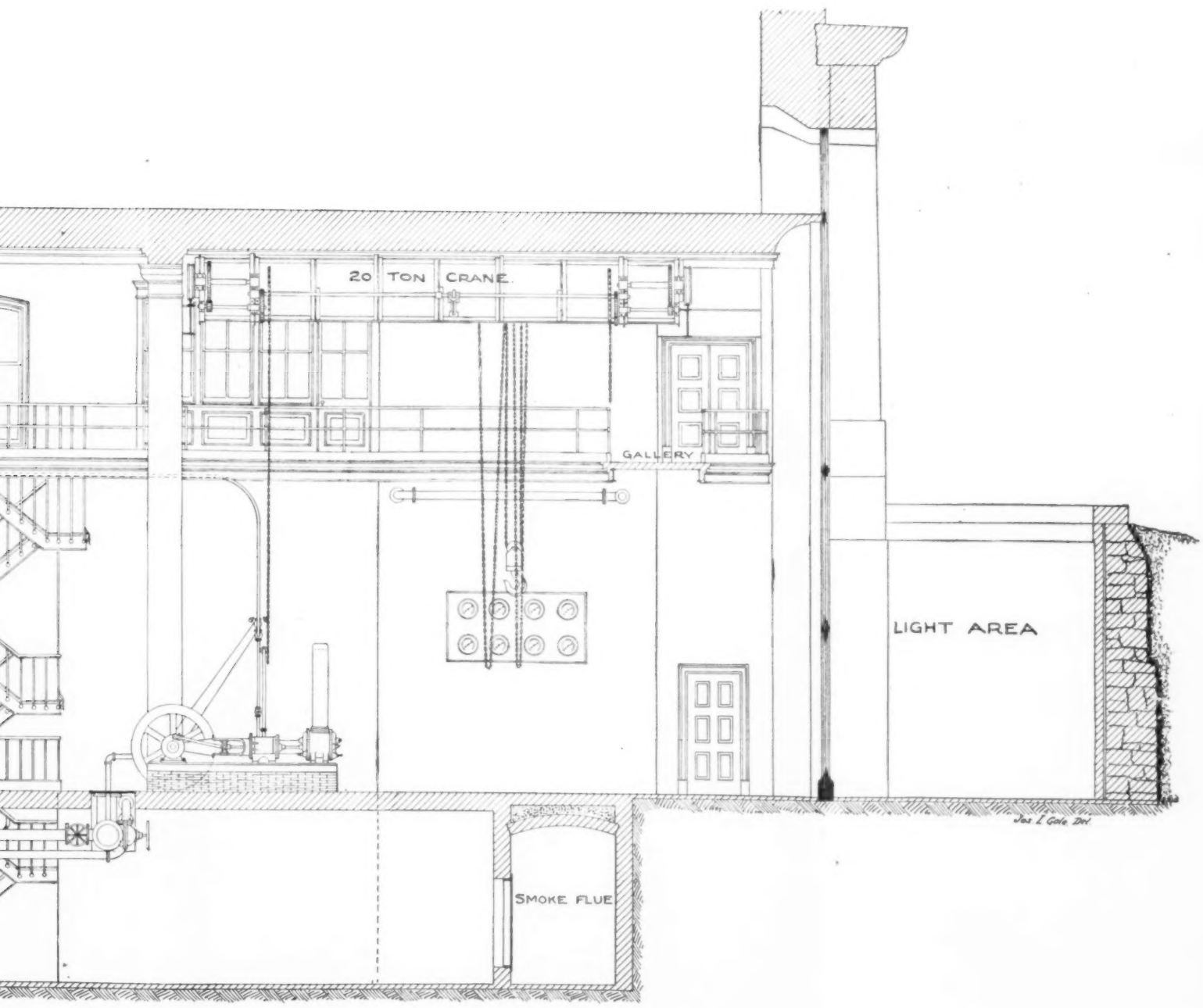


LONGITUDINAL SECTION THROUGH ENGINE AND BOILER ROOM LOOKING SOUTH.



ON THROUGH BOILER AND ENGINE ROOM LOOKING NORTH.





HEAT, LIGHT AND POWER PLANT
COLUMBIA UNIVERSITY
116TH STREET SITE.

and the two old buildings which were to be made use of, amounting in all to about one-half of the ultimate service.

The plan in Fig. 241 shows, in outline, the power house, with the coal vaults and the distributing tunnels.

Figs. 242 and 243 show a foundation plan of the power house, showing flues, air and ash ducts, etc., and this is followed by sections taken north and south and east and west, through the power house (Figs. 244 and 245).

The upper tunnel leads south from the floor above the power

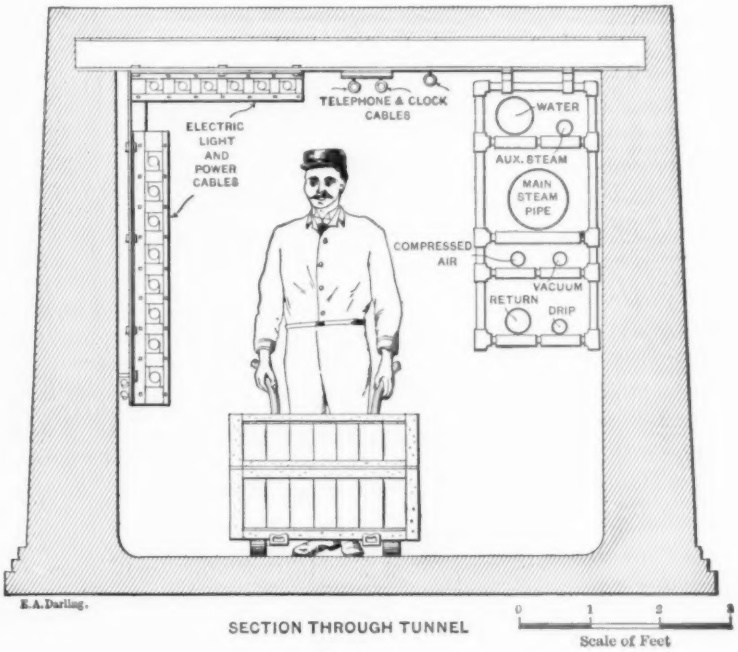


FIG. 246.

house to the library. Those on the lower level lead east and west to the vaults on the level with the basements of the outer line of buildings, and open into a gallery that runs entirely around the room 19 feet above the floor, and opening out on to the transverse roadway, which runs through University Hall and separates the power house from the gymnasium.

As the tunnels had to be built in filled ground, it was necessary to allow the filling as much time as possible to settle, then cut down trenches for the tunnels, sheet-piling the sides, build

the tunnels in the open and fill in again as quickly as possible, to allow a final settlement before putting down the court pavements overhead.

A section of the tunnels showing the pipe and cable carriers is shown in Fig. 246.

The coal vaults, of which a section is given in Fig. 247, have a capacity of 4,000 tons, sufficient to supply the boilers during the

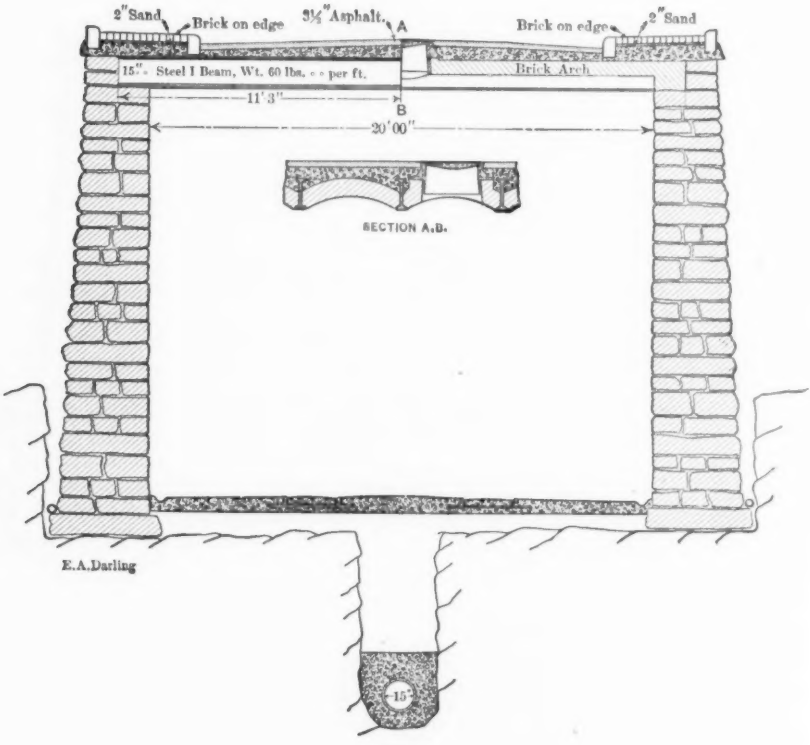


FIG. 247.

ten weeks of severe winter weather, when the river is practically closed to navigation. Coal is dumped down through holes that occur at regular intervals along the driveway. No trimming is necessary, and it is served to the boilers as required on a system of Hunt tracks, that are given a slight down grade all the way to the boiler room.

Water and gas mains are run in from the street on either side, and are hung up in the upper corners of the vault, where they

are always open to inspection as the flow of coal leaves the corners unoccupied.

Reference to plans and sections of the power house will show that the boilers, which have the maximum length of tube used in this type, are in two rows facing each other, with just sufficient room in the alley to draw a tube and clear the gauge glasses in the opposite row. North of the boilers and in the same compartment with them stand all the pumps in a row, and in the alley north of the boiler room, which connects the coal vaults together, stand all the tanks, filters, feed heaters, etc., each of these units being as close as possible to the pump with which it does business. These tanks were all fattened, flattened, or attenuated, as the case might be, to make them line up and give gangway for the coal cars.

The engine compartment is separated from the boiler room by a glass and iron screen which only runs up to the level of the gallery, making the entire room open overhead.

The five generating engines stand on a common foundation, which is simply a mass of rock and concrete of 1,000 tons weight, separated from the foundations of the building on the south and east by a smoke flue 10 feet deep, on the north side by the passage which leads to the base of the east chimney for the removal of flue ashes, and on the west by the cross tunnel which connects the ash tunnels with each other and with the elevator.

A 25-ton travelling crane runs north and south directly over the flywheels of the large engines, and the areaways opening out of the engine room and boiler room are of such a width that no difficulty is experienced in taking in or out any part of the plant. The entire framework of the windows, which are 12 feet wide and extend the full height of the room, may be removed for this purpose, as it is of iron, bolted together in sections. Part of the area in the engine-room side is covered at a height of 10 feet with a skylight, and forms the engineers' toilet and locker room. It contains toilet accommodations, shower baths, and lockers for the employees.

The boiler room is paved with brick all around the boilers, and with tiles around the pumps. The engine room is paved with the same buff-colored tiles, which are sufficiently strong and thick to permit the dropping of a wrench or hammer without cracking. Trench covers are of thick slabs of slate, with

hook holes in each end for lifting. The walls are of Keene cement painted and enamelled, and there is a wainscot around the entire room on the main floor, and again on the gallery level, 6 feet 6 inches in height, of white enamelled brick, all corners being rounded. The wheel pits are lined with enamelled brick, and the boiler settings are faced all around with moulded enamelled brick to their full height and capped with dressed blue-stone. The gallery has a steel framing and floor plates filled in with ash concrete, which is covered with a wearing surface of marble terrazzo.

The stairways have slate treads, and the railings of both stairways and galleries are of copper with a bronze hand rail.

The steel columns which support the cross walls and floors of the building overhead are encased in sheet-steel shells filled in with loose asbestos, to comply with the New York building law, although the only wood in the room is in the doors.

To those to whom the treatment of wall and floor surfaces may perhaps seem extravagant, we would say that the object in view was not to strain after ornamental effect, but simply to secure cleanliness and light, just as one would seek these ends in the kitchen of a modern residence or club. It is not so much a question of manners as of morals.

Boilers.

There are in all 17 Babcock & Wilcox all-wrought-steel boilers, having a total grate area of 825 square feet and a total heating surface of 37,000 square feet, built for a working pressure of 150 pounds to the inch; at the present time 110 pounds is the working pressure. All exposed parts of the steam drums are clothed with magnesia covering, to keep down radiation.

Coal is brought from the vaults in charging cars which pass over a track scale to give the tally, and line up within easy firing distance of the furnace doors. McClave shaking grates are used, and the ashes fall into hoppers underneath.

Having in mind the tendency of these hoppers to warp from the heat given off by the masses of hot ashes, and the radiation from the grates overhead, as well as the rusting effect from the water column blow-offs and the ash-pit sprays, they are made of $\frac{1}{4}$-inch steel plates riveted to angle-iron corners and lined throughout with firebrick laid in Portland cement mortar. Visor valves at the bottom of the hoppers dump the ashes into ash

cars, which are then run around to the engine-room elevator and lifted up to the level of the ash chute, which in turn empties the ashes into carts that pass through the driveway. The bodies of the ash cars are made to raise, tilt, and discharge over the truck wheels into the chute by the turning of a hand wheel at the side of the truck. As the ashes are thoroughly wet down when the fires are cleaned there is no dust, and one man attends to the whole operation of getting out ashes.

A yearly contract is made with a concern that agrees to cart the ashes away without cost to the university, and sells them to building contractors for the making of light concrete.

Sweepings and rubbish from the buildings and grounds are brought down in bags and burned under the boilers. As the carting away of ashes is at best a mussy performance, it is required to be done by eight o'clock in the morning, before the college day begins.

As the people of New York are restrained, not only by their desire to be free from the smoke nuisance but by the regulations of the Board of Health, from using soft coal, they are limited to the use of small sizes of anthracite for steam fuel. It is our belief that as long as the price of the best grades of this coal keep within reasonable bounds, the greatest economy lies in the use of the most expensive grades; but if through a combination of dealers, or a growing scarcity in these grades, the price is forced up, we have prepared ourselves to handle the lowest and most refractory grades of coal that come to market, by installing a system of forced draft, that stands ready to help along combustion whenever needed.

The air chambers run along between the smoke flues and ash tunnels, and are divided from the smoke flues by as thin a wall as practicable, to the end that the incoming air may borrow as much heat as possible from the outgoing gases.

A direct-driven Sturtevant blower is set in one end of the air tunnels, and the air is admitted through gates opening directly into the top of the ash hoppers under the centre of the bridge wall.

The gases of combustion escape down back of the rear headers, which are protected from them by a removable sheet-iron screen, through dampers set in holes in the roof of the flue. The smoke flues are 10 feet in height by 6 feet in width, arched and cross-arched with an air space between to avoid overheating the floor

of the engine and boiler room under which they pass on the way to the chimneys.

The chimneys are of steel plate varying in thickness from $\frac{5}{16}$ inch at the bottom to $\frac{3}{16}$ inch at the top. Each chimney is anchored at the bottom to a rigid framework of 15-inch steel beams, which is built into the shaft enclosing the chimney at the level where the top of the flue opens into it. As we had to follow the piers of the proscenium arch of the theatre up through the building, and were thus limited in width, it was necessary to flatten the chimneys after the manner common in steamship practice, to get sufficient area.

They measure 6 feet one way and 8 feet the other, and have a height of 135 feet above the grates.

A lining of thin cast-iron plates for the first 25 feet in height protects the chimney shell at the uptake.

Curb rings are built into the shaft at the level of the first floor of the building, and again at the attic floor with six ball-pointed set-screws which barely touch stiffening bands on the chimney to steady it, and allow it free movement up and down for expansion and contraction.

The reason for using such heavy plate in the chimneys was to insure long life against corrosion.

It will be noticed that while the chimneys have an area of 40 square feet, the smoke flues, which are tapered by giving the floor of the flue a pitch toward the stack, have an area at their greatest section of 60 square feet. This excess of area was given to allow for the deposit of flue ashes on the floor of the flue, and for drifts of ashes which were liable to form at the turns. As a matter of fact, during the first season's run a drift 3 feet in height formed, and had thus cut down the area at that point to 42 feet.

There is a large pipe chase built clear to the roof next each chimney shaft; that on the east side being for the free-exhaust riser and its return drip, and that on the west side to contain the twin standpipes for the water system. The chases are of sufficient size to allow for lowering a man to make or unmake the flange joints in these lines.

The feed-water and blow-off mains run through the ash tunnels, and are visible and accessible throughout their length. The blow-off line and its connections are of heavy cast iron, and the feed line is of drawn brass.

During the winter season the boiler feed is served from the return and low-pressure drip tanks, and is sent through a horizontal Wainwright heater.

When the weather is warm and the exhaust is being wasted, the boiler feed is taken from the swimming pool under the gymnasium directly through the feed heater.

The pool contains 250,000 gallons of water that has been first run through two tandem filters to cleanse it, and then brought to a temperature of 70 degrees Fahr. in a heater. The feed losses during the winter season are made up from the same source.

This scheme prevents the waste of either heat or water, and secures not only a constant change of water in the pool but clear water for the boilers. It works like a charm.

Engines and Generators.

This part of the plant was divided into six units, as follows: One 24 by 48-inch, 600 horse-power, simple Corliss engine driving a 400-kilowatt generator at 90 revolutions per minute; two 20 by 42-inch 300 horse-power simple Corliss engines, driving 200-kilowatt generators at 100 revolutions per minute; two 11½ by 16½ by 12-inch cross-compound, high-speed engines, driving 75-kilowatt generators at 260 revolutions per minute; one 50 horse-power gas engine, driving a 30-kilowatt generator. All engines to be direct connected to their generators.

The larger engines were intended to bear the heat and burden of the day during term time from eight in the morning until seven at night, the high-speed cross-compounds to take care of the light loads in the early morning and after seven at night, and the gas engine was to be put in to furnish the comparatively small amount of current called for during the summer vacation, and allow for shutting down the boilers entirely for three months in the year. Our specification for this unit called for an engine that would tend to business like a steam engine, and it must neither lie down when pushed nor cough all day long.

We will, no doubt, get one in time.

As the writer is a firm believer in the proposition that standard commercial types, representing the average sense of all, are the best types to select from when you are buying, the only vagaries he indulged in when specifying the make-up of the

Corlisses was the insertion of one clause which called for an extended bed-plate which should run around the flywheel, take hold of the field ring of the generator and grip the pillow block,

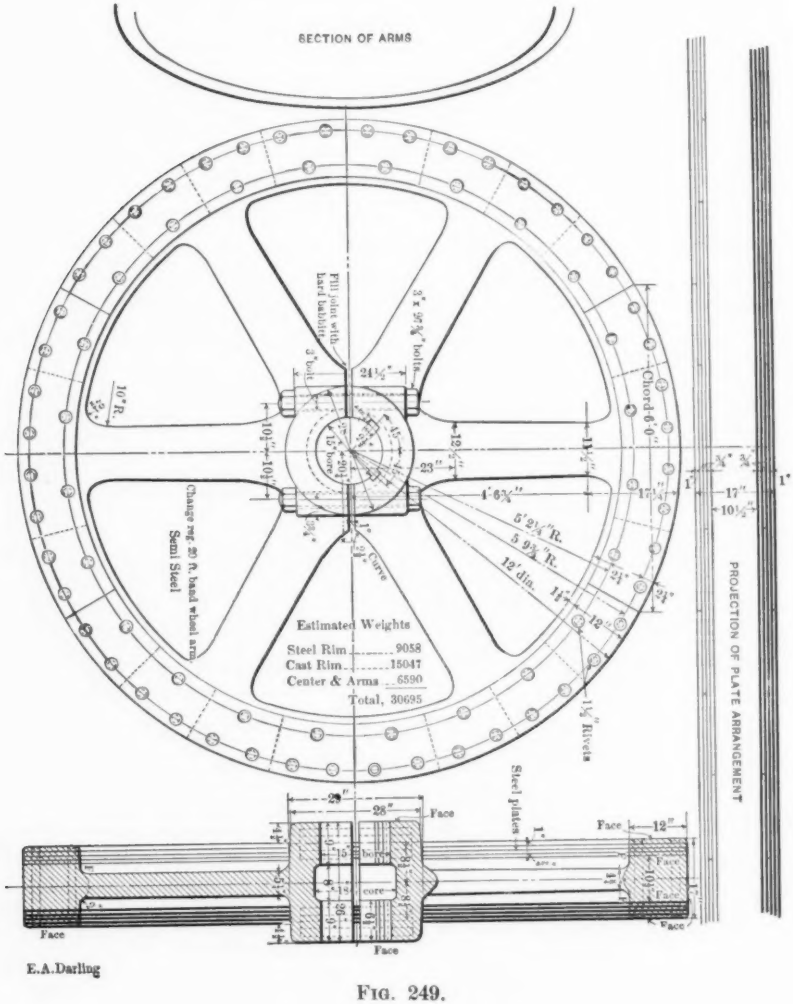


FIG. 249.

the idea being to make a unit of the whole machine and insure permanent alignment after the manner already common in high-speed practice.

The other clause called for a flywheel that would stay put, no matter what the rest of the engine did, in case it should run

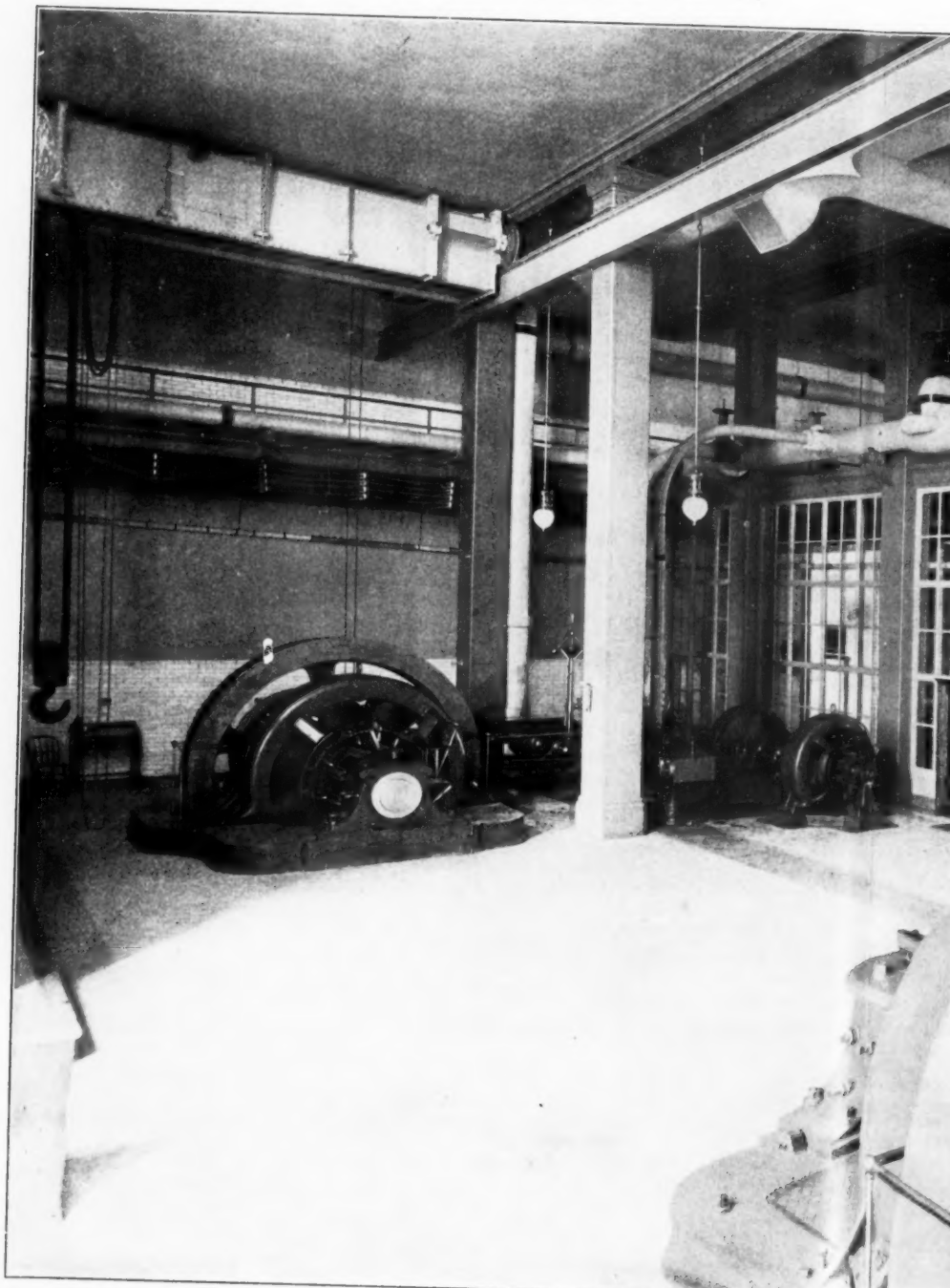
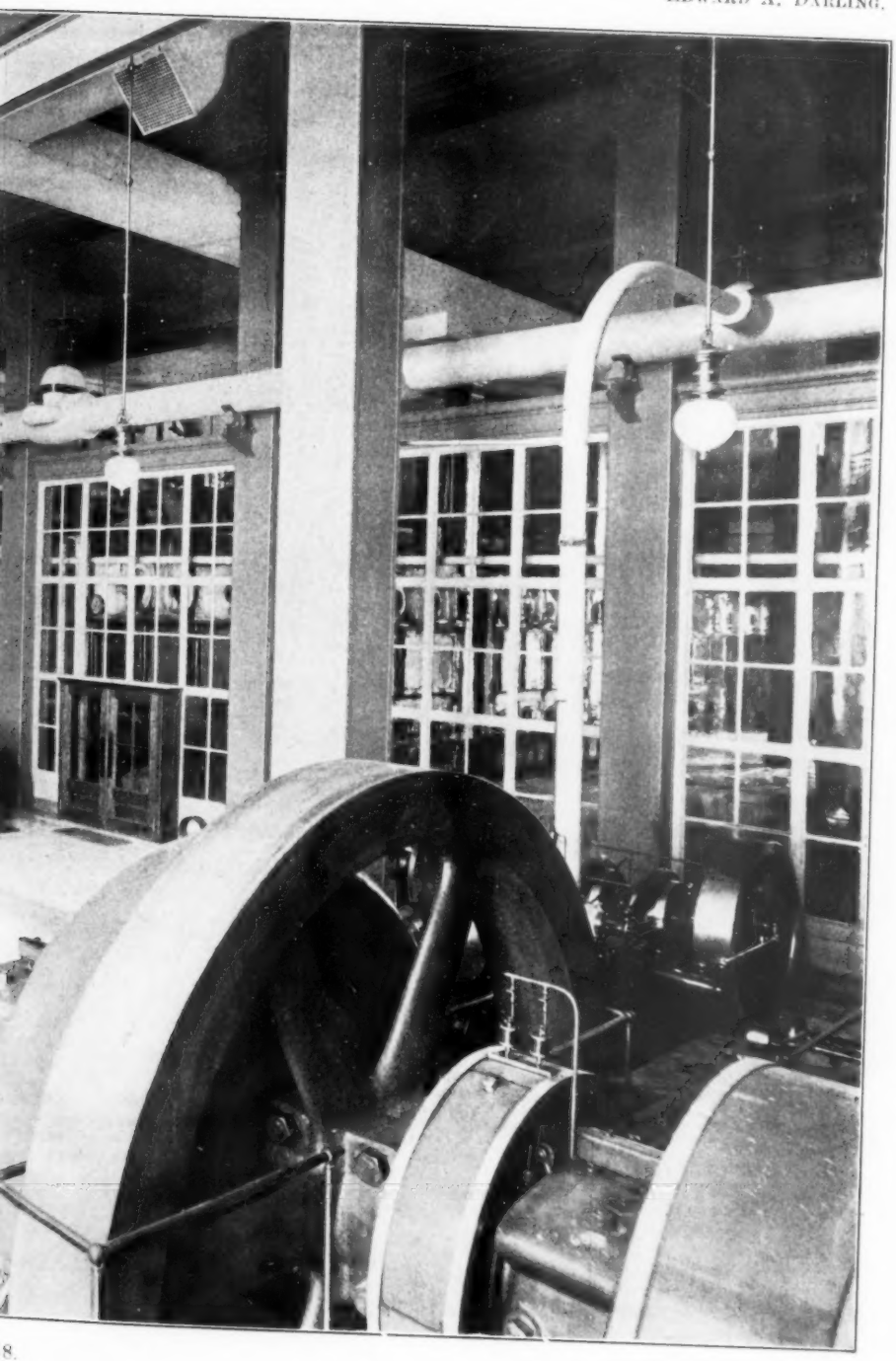


FIG. 248.



8.

amuck, and it must run true even if it had to be turned in place.

The way these modest requests were met by Mr. Edwin Reynolds, of E. P. Allis & Co. (to whom the writer here makes grateful acknowledgment), is shown in the following drawings

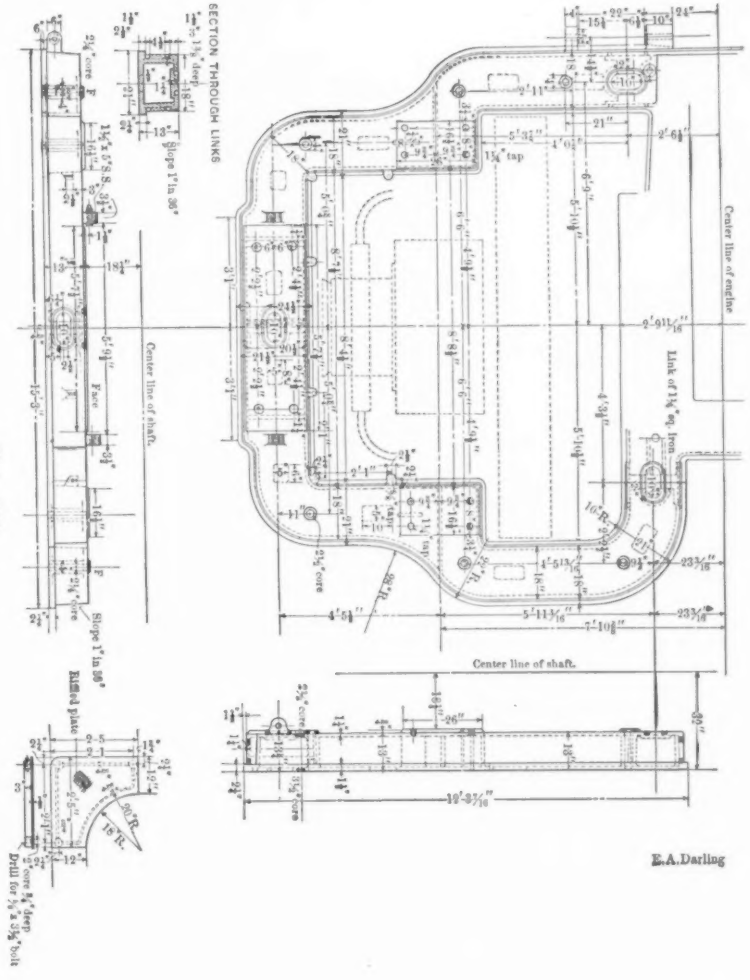


FIG. 250.

and photographs of the 300 horse-power units which were first installed (Figs. 248, 249 and 250).

Each flywheel is 12 feet in diameter and weighs 15 tons; the body of the wheel was cast in one piece, with the hub parted to allow for shrinkage in the arms. After the rim had been faced

off on both sides, four layers of mild steel segmental plates having a total thickness of $3\frac{1}{4}$ inches were laid up on each side, with joints staggered, 2-inch holes drilled through and countersunk, and turned soft-steel pins driven through and riveted cold. When this operation was completed, the wheel was placed on the boring mill, the rim finished all over, and the hub bored to gauge. Shipment from Milwaukee was made on a flat car that had a slot cut in its floor, which allowed the rim of the wheel to come down within a foot of the ties as it rested on a rough oak gudgeon which took the place of the shaft and rested in turn on timbers laid along the floor of the car.

In the rooms directly over the engine room there will be at times as many as twenty-five hundred people. Due regard for this fact led us to put in an additional safeguard against the danger from a racing engine, in the form of automatic stop valves above the throttles of the engines, a precaution as necessary, in our opinion, as putting a safety valve on a boiler.

It will be noticed that these engines are very "short in the barrel," as a horseman would say; the armature of the generator butting up against the hub of the flywheel so close that there is barely room to put your hand between. This was done to secure compactness and rigidity, although there would seem to be small chance of springing, with a shaft 15 inches in diameter for an engine of this size. The complete unit weighs 65 tons.

The high-speed cross-compounds were built by Armington & Sims, who put in our first electric-light engine at Forty-ninth Street in 1886.

These engines are set in the bight of the larger engines, with their throttles within reach of the throttles of the larger engines, so as to be handy when running in parallel. As the type is doubtless familiar to the members of the Society no description is necessary.

At the solicitation of the writer, who makes no claim whatever to proficiency in electrical matters, Prof. F. B. Crocker, past president of our sister society, the American Institute of Electrical Engineers, consented to write specifications for generators that would stall the engines before they could burn out the armatures or cause a sputter at the commutator. As the bill of particulars is brief and to the point it will bear setting down here, and should be of interest.

Specification for Dynamos for Columbia University.

The manufacturer shall furnish and install in the power house of Columbia University ready for service:

Four (4) direct-current, compound-wound dynamos, to be directly connected to the shafts of the engines, outboard bearings and all foundations to be furnished by the engine contractor.

Capacity.—Two of these machines, which must be duplicates of each other in every respect, all parts being interchangeable, are to be directly connected to two Corliss engines.

Each dynamo shall have the following capacity:

Output in kilowatts	200
Volts at no load	118
Volts at full load	124
Ampères at full load	1,600
Revolutions per minute	100

The other two machines, which must also be exactly alike, with all parts interchangeable, are to be connected to high-speed compound engines, and shall have the following capacity for each:

Output in kilowatts	75
Volts at no load	118
Volts at full load	124
Ampères at full load	600
Revolutions per minute	260

Regulation.—These machines shall be wound to overcompound from 118 volts at no load to 124 volts at full load, with a constant speed. This is intended to counteract a 2 per cent. decrease in speed of the engine and a 3 per cent. drop on the conductors at full load. This rise in voltage shall be approximately uniform—that is, not more than 3 volts nor less than 2 volts at half load, variations in speed being prevented or allowed for.

There must be sufficient range in the series winding to allow the overcompounding effect to be varied from 3 per cent. to 7 per cent.

The shunt windings shall be excited from the bus bars, and must have sufficient range to enable the voltage to be regulated up to 130 volts or down to 110 volts at full load and normal speed.

Any or all of these machines must operate well in parallel, each taking its share of the load (within 10 per cent.) in proportion to its size.

Efficiency.—The commercial efficiency of these machines, when hot, shall not be less than the following:

Full load	91
Three-quarter load	91
One-half load	86
One-quarter load	80

These figures are intended to secure a good efficiency at all loads of one-quarter or more, rather than a specially high efficiency at full load with the sacrifice of efficiency at fractional loads.

Heating.—No part of the machines, after a continuous load of twelve hours at full speed, must be heated more than 40 deg. C. above the temperature of the surrounding air. After a continuous run for six hours at full load, and a further run of two hours at full voltage and 25 per cent. excess over rated number of ampères, no part of the machine must be heated more than 50 degrees C. above the temperature of the surrounding air. These temperatures are to be measured by a thermometer and by increase of resistance in the case of the armature conductors and field magnet. When the latter method is used the heating limits specified above may be increased 15 per cent.

Commutation.—A continuous run at full load for twelve hours must not produce any appreciable sparking, or any cutting, pitting, rising, lowering, or unequal wearing on the surface of the commutators; the brushes must not show evidence of burning or cutting beyond ordinary wear, and the commutator shall receive no attention except lubricating and wiping during the test.

A continuous run for two hours at 25 per cent. overload must not produce injurious sparking, cutting, or burning of the commutator or brushes.

Brushes.—Brushes must be so arranged that they can be tripped simultaneously when starting or stopping the engine.

Insulation.—The insulation resistance between the armature and field conductors and the frame of the machine shall not be less than one megohm when measured separately.

The insulation of the armature and field conductors shall be tested with a pressure of 1,000 volts (direct current).

Gauge for Armature Bore.—A plug gauge for the armature bore will be furnished by the engine contractor.

The armature must be perfectly balanced and must not run out of true more than $\frac{1}{64}$ of an inch either axially or radially.

Finish.—All castings to be filled and rubbed to a smooth finish.

Erection.—To be done by the builder, who will furnish all labor, materials, and superintendence.

Generators to be delivered ready for connection to feeders to switchboard.

Delivery.—Generators must be in place not later than July 1, 1897.

Finally.—All materials and workmanship are to be guaranteed of the best, and will be subject in detail to the approval of the superintendent of the university or his authorized representative.

After securing competitive bids from some of the principal concerns in the business, the contract was awarded to the lowest bidder, the Crocker-Wheeler Company.

The greatest compliment I can pay the generators is simply to say that they are in tune with Mr. Reynolds's engines. And that is no faint praise.

A brief description of the method of preparing the engine foundations may be of interest.

One of the most troublesome problems encountered in putting heavy machinery under a building that is devoted to uses entirely foreign to the generation of power is to get rid of the annoyance caused through the transmission of vibration by the bearing walls and piping. This becomes especially difficult where the foundation rests on bedrock, as in the present instance.

The attention of the writer was called some years ago to a series of very interesting experiments made by the Japanese Government where plans were formulating for the buildings of the University of Tokio. The frequency of seismic disturbances in that region was a constant menace to the integrity of masonry structures, and the experiments showed that the vibration was greatest on the surface of the ground, diminishing rapidly as lower levels were reached in pits dug for the purpose. The seismographs used for the purpose showed in addition that the vibrating waves followed horizontal lines around the earth's

surface. In the light of the knowledge thus gained, the building walls were kept clear of the surrounding ground by areas that extended all the way around, cut down to a level just above the line of footings. Comparatively severe earthquakes that have occurred since the buildings were erected have been scarcely felt within the buildings, and have occasioned no damage whatever to the structures.

A simple way of noting the direction of vibrations caused by the impact of a blow on a resisting surface is to have some one strike an anvil while you move your hand up and down over its surface. A blow on any part of the face of the anvil is plainly felt at any other part of the face, while the base gives out no appreciable response to the blow.

It seemed to us, then, that the best way to avoid giving continual annoyance to those who were to occupy the upper part of the building was to unite all the engine foundations on a single block of masonry, cut clear of the building foundations, that would be in effect an anvil block of sufficient mass to absorb the vibrations before they reached the level on which the building foundations stood.

This was tried first when we built the medical school engine room, and it worked so well that it seemed worth repeating in the larger scheme. An added difficulty was met here, however, on account of having to take care of the row of columns that ran across the engine-room floor, and of having to carry the surface of the floor across the top of the smoke flue, which served as an insulating trench next the outside walls.

For the columns, we borrowed a scheme that had been adopted at the Houston Street cable plant, cutting pits in the rock for the bases of the columns to rest in, then boxing in the bases to the floor level and filling up against this with concrete.

In order to baffle the vibrations that might travel across the arches over the smoke flue, we laid up soft-pine planking on edge against the wall to the depth of the arching, and finished the floor up against it. While this lasts it forms a deadener, and when it rots it will leave an air space.

To prepare the foundation block for the engines, the entire area was cleaned of all loose or rotten rock, the wheel-pits and exhaust trenches were laid up in brick to the working level, and the iron conduits set in place for the feeder mains from the generators to the switchboard. The whole area was then

brought up to the working level with Portland cement concrete, and the surface floated off.

After the mass had hardened sufficiently to admit of drilling, the templates for the different engines were laid down and the holes for the anchor bolts marked out. The holes were cut with a steam rock drill, and were made an inch larger than the bolts they were to receive. Those for the larger engines were sent down 10 feet, and for the smaller engines 4 feet.

The bolts, which had been roughed by the blacksmith all the way down the shank, were then dropped into the holes, and the engines were set up without any of the usual cramping and fidgeting caused by set anchor bolts, as the bolts were free to wabble about and accommodate themselves to the inevitable variations from template that occur in a rough bed-plate. After getting the engines to fair and true level by means of stake wedges the bolts were given a full nut, the holes grouted to the brim with Dyckerhoff cement, and the joints finally rusted. Those bolts are there to stay.

When all the engines are running no vibration can be detected on the engine-room floor or in the walls of the building, but the framework of the gallery, we are sorry to say, seems to borrow a little tune from the pipes it carries, which is transmitted to the steel structure above the engine room. We are thinking about this all the time and hope to cure it. Suggestions would be welcomed.

Piping.

Simplicity, brevity, and elasticity are of the very essence of good practice in this line, and we believe that these ends should be sought before all others in laying down a plant. It is better to make the engines and boilers fit the piping than to go the other way about it. An unnecessary turn or length of pipe occasions a never-ceasing waste from friction and radiation.

Considering the piping as just so much machinery, the next most important points to provide for would seem to be:

1. To put it up so that it might adjust itself freely under the strains imposed by expansion and contraction.

2. To consider the human element involved in its operation, by setting all valves where they can be easily and quickly handled without making undue calls on the heroism of the engine-room force in case of an emergency.

3. To put the piping together in the way you would any other machinery, with bolted joints that can be easily made and unmade without destroying or damaging either pipe or fittings.

Finally, to treat exhaust steam, water, and compressed air with as much consideration as you would live steam, remembering that the same laws apply to them with perhaps greater force in that they may have less energy to overcome obstacles in the shape of tortuous pipe bends, crooked valve passages, or the strictures imposed by reduced areas.

Our performance along these lines falls often short of our ideals, but we have at least had the satisfaction of trying to live up to an ideal.

A 16-inch header, in the form of a flattened ring, runs around the outside of the boilers, and rests on roller bearings that are carried by the gallery.

The boilers feed into the inside of the ring, and the engines and distributing lines connect directly to the outside of it, the connections to both engines and boilers being single, straight-away, long-turn bends. The angles at the turns of the header are formed of steel castings.

A 14-inch exhaust main runs in a trench straight across the engine-room floor, parallel to the steam main overhead, and the cylinders of the engines fairly empty into it. It will be noted that there is nothing but the valve between the exhaust nozzle of the high-speed engines and the main.

To allow for movement in the rigid connection which this forms between the engines an expansion joint is placed in the centre of the line, and a screw joint was made to the exhaust nozzle of the Corlisses on the end. As the total distance from the steam header to the exhaust main is only 25 feet, we come pretty near getting boiler pressure at the throttles and a minimum back pressure at the exhaust.

The pumps and other auxiliaries are given the same chance, by running their steam cylinders along in a line over a branch exhaust main, that opens at right angles into the large main, at the point where this main branches to the feed heater and the heating system of the building. Just outside the engine, to the south, the main rises to the level of the library tunnel and continues along it, full size, to the heating stacks of that building.

There are separate systems for the high-pressure and low-pressure drips; the high-pressure being sent back directly to

the boilers, and the low-pressure returned through the return tank and feed heater with the regular feed. The high-pressure lines were to be dripped at every fitting on horizontal runs.

Our specifications and drawings called for flanged connections on all pipe over 3-inch; for outside screw and yoke Chapman gate valves, of varying weights on all sizes over 3-inch; for long-turn elbows wherever possible; for nothing less than standard full-weight pipe, no matter what purpose it might be intended to serve; galvanized pipe and fittings for cold-water lines, drawn brass (iron-pipe sizes) for feed lines, and cast-iron for blow-off lines. A choice between Johns asbestos and Keasbey's magnesia was allowed for pipe covering.

Auxiliaries.

Two 9 by 5¼ by 10-inch duplex pumps alternate in the service of the boiler feed.

Two 8 and 12-inch by 10¼ by 10-inch compound duplexes controlled by pressure regulators take care of all the water supply, with the exception of the fountains in the south court, which are to be operated on a continuous circuit by a 300-gallons-per-minute electric pump placed in the basement of the library building. A 6-inch Croton line was run across through the coal vaults, and connected on the one end to the city main in Amsterdam Avenue, and to the main in Broadway on the other end, both ends being kept open to prevent stagnation, and we are thus assured a constant supply in the event of either main being shut down.

The water flows through the filters, under street pressure, into the suction tanks of the house pumps.

Next in line are two 14 and 20-inch by 10¼ by 10-inch elevator pumps, which operate three elevators in University Hall.

The elevators in all the other buildings are electric, with the exception of the Otis experimental elevator in the engineering building.

A 20 by 12 by 10-inch fire pump, figured to throw 1,000 gallons per minute, delivers directly into the house mains and risers of the inside buildings, these lines being made unusually large on this account.

The suction tank for the fire pump is the swimming pool aforementioned. No danger of a water famine for the fire

pumps. To keep in practice, this pump is used for emptying the pool into the sewer every two weeks, when the pool is scrubbed.

Compressed air is distributed to the buildings where it is needed from a Rand compound compressor, with steam cylinders 9 and 13 inches, a stroke of 13 inches, and air cylinders 12 inches in diameter.

A 9 by 12 by 10-inch wet vacuum pump runs all day long to serve the requirements in that line of the chemists and physicists in their lecture rooms and laboratories.

To give an idea of what these side issues amount to, we need simply say that in Havemeyer Hall alone there are over 6,000 outlets and connections for water, compressed air, gas, and vacuum.

Electric Distribution.

The switchboard is divided into three panels; the central panel carrying the generator switches, circuit breakers, and instruments, one of the side panels carrying the "power" connections for the buildings, and the other the "light" connections.

The whole board is 40 feet in length, and was designed by Mr. Mailloux, to cover the entire group of twenty buildings, without disturbing the bus bars and switches for the buildings first erected, as buildings are added from time to time.

Although the generators ordinarily carry the light and power load together, the circuits are kept separate throughout, and, with the steady load furnished by the ventilator motors, the throwing on and off of elevator and shop motors does not interfere with the regulation to any appreciable extent.

Whenever there is a tendency to jump big loads on and off the circuit, as, for instance, in the electrical laboratory, and more recently the industrial chemistry and metallurgical laboratories, with their new-found delight, the electric furnace, we have put circuit breakers in the power lines, where they will localize the annoyances incident to short-circuiting to those who may be indulging in pyrotechnics. A local circuit for alternating current is provided in the engineering building, operated by a transformer taking current from the power circuit, and when the use of electric furnaces becomes more extended we shall put in a transformer in Havemeyer to facilitate work in this line.

The feeder mains terminate at local switchboards in the base-

ment of each of the buildings, which divide the circuits by floors, and panel boards in the hallway on each floor subdivide the service to the separate rooms.

This method of distribution gives local control of the circuit in so far as it is practicable and makes the operation of lines a comparatively simple matter.

Every switch or cut-out is lettered or numbered to correspond with a schedule or diagram posted beside each board.

From the main switchboard to the branch boards the feeder-main cables are carried by segmental porcelain blocks, fitted into steel trellis frames which are hung under the power-house gallery or bolted to the side walls of the tunnels. Over 65 tons of copper were put into the mains for the buildings now in use.

In addition to the wiring for light and power there are complete interconnecting systems for telephone, electric clock, and signal bells, and for the watchmen's patrol record within the buildings. These wires are run in iron conduits like the light and power circuits, and connect in the tunnels to lead-covered cables.

There is no reason why the electrical installation throughout should not last as long as the buildings, as it is made up in the most permanent and enduring manner, and the table (Fig. 251) gives a general idea of the ultimate electrical requirements of all the buildings at the new site.

The appended schedule gives details of the electric feeders now in place.

SCHEDULE OF ELECTRIC FEEDERS.

No. of Feeders.	Terminates at Building –	Maximum Probable Load, Ampères.	Length in Feet One Way.	Loss in Volts.	Size of Wire, Circular Mils.
		LIGHTING FEEDERS.			
1	Library	1,100	376	2.00	4,000,000
2	Engineering	525	669	2.50	2,700,000
3	Havemeyer	566	544	2.50	2,590,000
4	Fayerweather	244	409	2.50	800,000
5	Schermerhorn	404	291	2.50	900,000
6	University Hall	126	125	1.50	211,600
7	University Hall	422	125	1.10	1,000,000
		POWER FEEDERS.			
8	Library	660	398	5.00	1,000,000
9	Engineering	350	700	4.20	1,200,000
10	Havemeyer	855	570	5.00	1,900,000
11	Fayerweather	351	416	5 00	600,000
12	Schermerhorn	640	298	5.00	800,000
13	University Hall	980	174	3.50	1,000,000

Performance.

A log is kept, covering every day in the year, giving for each week the total hours' run of each unit, and the output, together with averages and the consumption of coal, water, and supplies

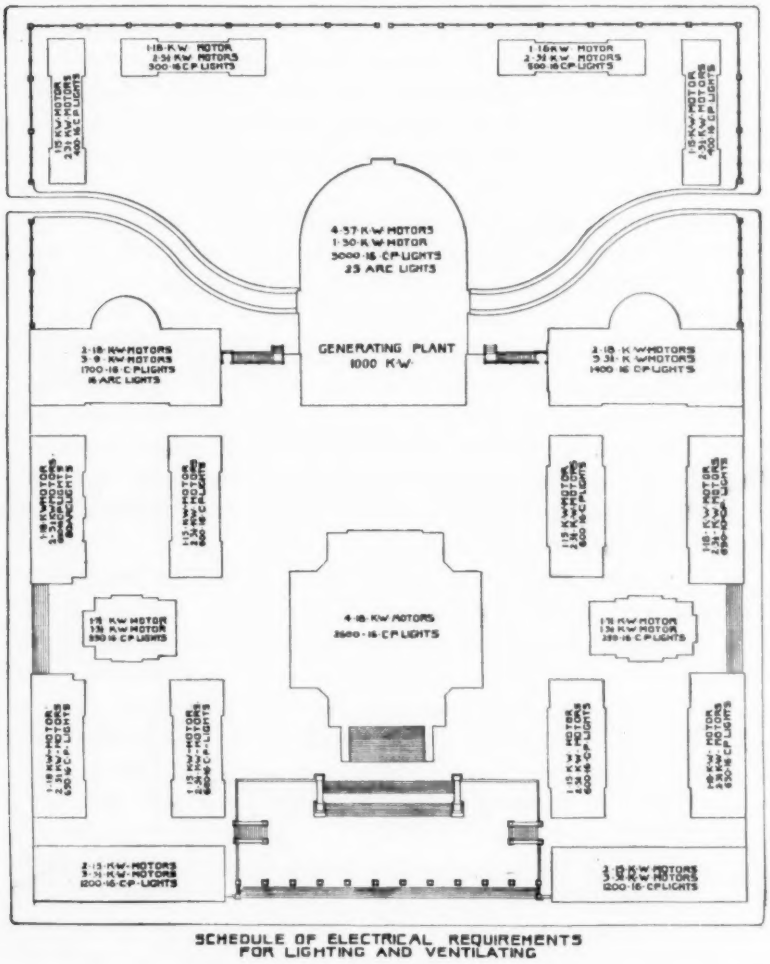


FIG. 251.

and the cost of labor and repairs charged against the power house.

Of course, aside from the desire to have a record of operations, the principal reason we have in keeping the log is that we may

know just how much money we are spending and what we are getting for it.

A sample sheet from the log-book is shown (see table facing).

To operate the plant as it stands to-day, during the winter months we employ the following force :

A chief engineer, two assistant engineers, two oilers, one dynamo tender, three heating-service men, one electrician, one machinist, four firemen, two coal passers, and one helper.

The heating-service men patrol the heating and ventilating plant in the different buildings at regular intervals day and night during cold weather; and considering the fact that there are in all, so far, 22 motors and fans ranging from 10 to 50 horse-power each, 21 return pumps, and an aggregate of 80,000 square feet of heating coils and radiators scattered throughout the buildings, it will be seen that they are kept pretty busy. The electrician attends to keeping all electric conduits and fixtures in order, and is allowed extra help as occasion demands.

The power-house force stands two twelve-hour watches during the winter months, when steam is kept on day and night.

During the season when light and power alone are required and the force is reduced, the day men report at 7 and leave at 6, and the night men report at 12.30 and leave at 11.30 when the library has closed for the night. By this arrangement the entire force is on duty during the afternoon when the load is heaviest.

The cost of heating a building depends in great measure on the amount of ventilation furnished. In all well-ventilated buildings the heat losses are far greater from the outgoing current of vitiated air than by direct radiation from wall and window surfaces.

The loss from this source can be calculated exactly by measuring the difference in temperature between the incoming and outgoing air and the volume of air moved.

While this may properly be called a loss, it is in no sense a waste. The amount of pure air required by a human being under varying conditions is a matter of common knowledge, established on scientific authority, and he should no more be required to breathe impure air than to drink foul water.

For the very reason that it is more difficult to detect a lack of ventilation than a lack of heat or light, this point should receive the closest attention.

That it is often neglected, either deliberately, through the

practice of a false economy or through sheer ignorance on the part of those who design or those who operate public buildings, is evidenced by the fact that for one building you find inadequately heated you will find a score without any ventilation worthy the name.

We estimate that when our ventilating system is in full operation we use 50 per cent. more fuel than when we are simply heating the buildings and allowing the heated air to escape by natural draught through the flues and doors.

In no part of any building do we give less than six changes of air per hour, and in some lecture rooms and laboratories as many as twelve changes per hour.

More current is used for driving the ventilation fans than is used for light, and the ventilating apparatus is run continuously during the time when the lecture rooms and laboratories are in use. When all the fans are running they will move 1,250,000 cubic feet of air through the buildings every minute.

For all purposes, we burn at the present time about $13,000 worth of fuel per annum, with a labor account chargeable to heat, light, and power of $10,000 per annum, or at the ratio of 1 to 1.3. Ultimately we expect to have to spend not less than $30,000 a year for fuel and to increase our labor account to $15,000 a year, which will then give a ratio of labor to fuel of 1 to 2.

The complete power plant and its connections, exclusive of structural work or excavations, represents an investment of $200,000, or at the rate of $50 per horse-power. That part of University Hall occupied by the power house, including the coal vaults and tunnels, cost $350,000.

The heating and ventilating equipment and the electric wiring of the entire group will foot up to $550,000. This makes a total of $1,100,000 for the complete engineering equipment, exclusive of drainage and plumbing.

The buildings and grounds at the One Hundred and Sixteenth Street side of the university already completed stand on the books of the corporation at $6,500,000. The entire improvement is estimated to cost twelve millions, including the cost of the land.

From the figures given above it will be noted that the engineering equipment represents about 10 per cent. of the total investment, which is not excessive, considering the important part it plays in conserving the convenience, health, and comfort of those who occupy the buildings.

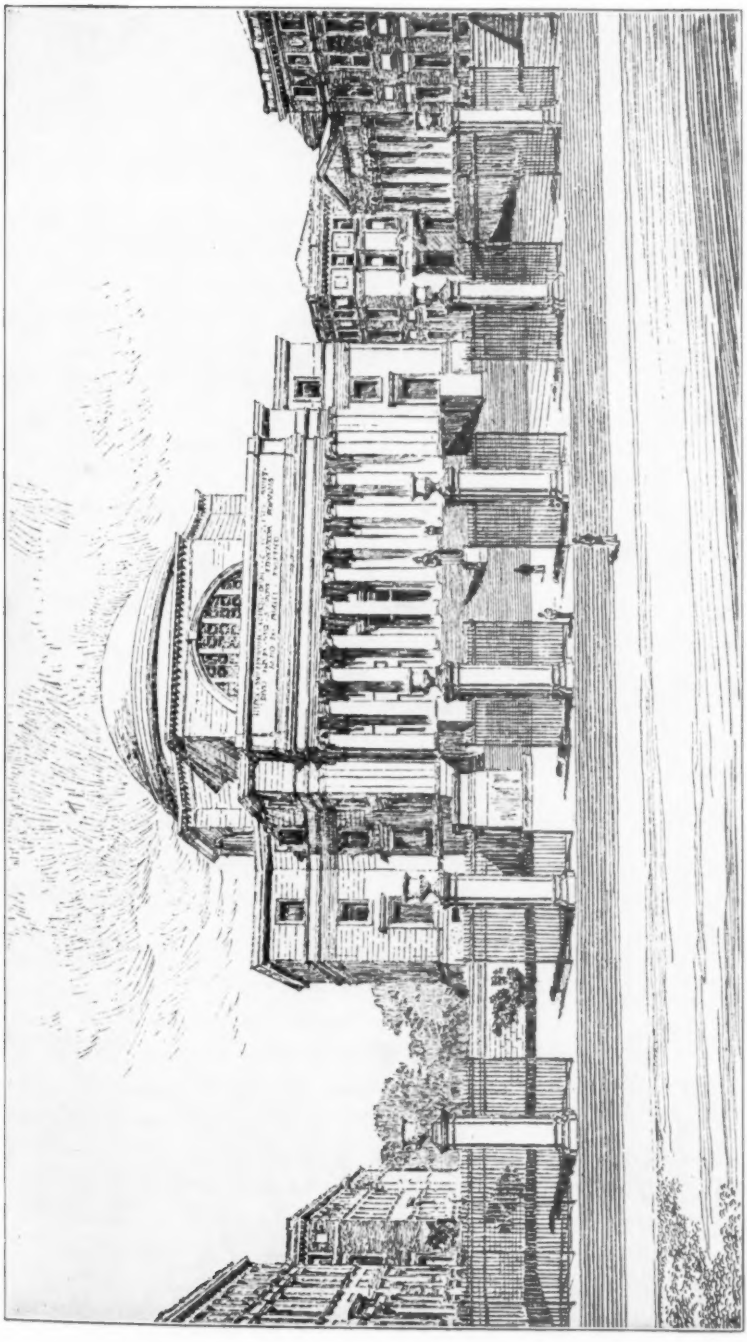


FIG. 252.

When the group of buildings on Morningside Heights shall have been completed, Columbia University in the city of New York, including all its schools and colleges, will have a plant costing not far from seventeen millions of dollars, with ample facilities for the 6,000 students and 600 teachers it will then be able to accommodate within its walls.

These figures appear large in perspective, but they are simply based on the ratio of increase maintained by more than one of our American universities during the past decade.

Of course the length of time it will take to realize this project will depend largely upon the character of the support given to the university by the citizens of New York and on the energy displayed by all who are interested in its welfare.

In conclusion, I desire to make acknowledgment first of all to President Low and the trustees of Columbia University, without whose countenance and support the development of the engineering plant on broad lines would have been impossible;

To the architects, Messrs. McKim, Mead & White, who said that the best in the engineering line was none too good;

To my co-laborers the consulting engineers, Mr. Wolff, Mr. Mailloux, and Messrs. Waring, Chapman & Farquhar, who gave the best that was in them, without regard to time or trouble;

To the contracting engineers, who waived all profit in many instances in order to help along what they were pleased to consider a good work;

To Professor Hutton, to whom I am indebted for valuable suggestions in the preparation of this paper;

And last, but by no means least, to Mr. F. A. Goetze, my able and conscientious assistant, whose resourcefulness in design and watchfulness in matters of construction I cannot praise too highly.

In presenting this paper before the Society, the writer has felt some diffidence on account of not knowing to what extent the members might be interested in a development of the kind.

He must rely on the well-known interest all engineers have for matters that concern education in its various phases, and on the indulgence of those whose grasp of the subjects dealt with may be based on a more extended knowledge and experience than his own.

DISCUSSION.

Mr. Reginald Pelham Bolton.—I would like to take the opportunity of personally thanking Mr. Darling for this very complete and full paper, which is very admirable in the details which it presents. It describes also a central power plant, and this is placed under very different conditions, as regards the surrounding buildings, from that described in the previous paper. In this case, I think there is no question but that the central power plant is the right one to adopt. Mr. Darling has made one remark in his paper on which he has also asked that the united wisdom of the Society be brought forth. On page 711 he mentions that he has some trouble with the vibrations of the pipes communicating to a portion of the building, and asks for suggestions on that subject. That has been a very troublesome subject in many of the tall buildings, and has been entirely gotten over by supporting the pipes from a different point. In this case the pipes are carried by the framework of the gallery, and it is not to be wondered at that it gets some of the vibrations. It may not be practicable to carry it from below. If so, I am almost afraid that the only remedy is to destroy the gallery, or take the pipes and put them somewhere else. The question of the vibration of the engine foundations has been dealt with in detail by the author in an admirable manner and represents the very best practice, by forming a very large mass of foundation isolated from the surrounding buildings.

In connection with the subject of fuels, which Mr. Darling has entered upon, I would say that so far as New York city is concerned I am heartily in favor of the use of anthracite coal. In the tall buildings anthracite is burned under ideal conditions, because we have an exceedingly strong draft without any cost, owing to the extreme length of the chimneys which are necessary to the buildings. On the other hand, even if that were not the case, I think that the interest of the whole city demands that we should burn a fuel which is smokeless, and that the Board of Health is entirely in the right in demanding that we shall not burn soft coal in New York city and in continuing its present crusade, which is pretty active at the present time, against the users of soft coal who emit any smoke from their chimneys.

Mr. Oberlin Smith raised a question of a remarkable fuel being used. I was told of another in Philadelphia as I passed through

this week. During the late blizzard, in one of the hotels there which depended entirely upon its own plant, having no stand-by from any other concern, they were unable to keep up the supply of fuel, and when the coal ran short they burned mattresses valued at $8 apiece.

Mr. William H. Bryan.—Mr. Darling tells us an interesting story, and, what is even better, tells it in a refreshingly interesting manner. There are a few questions, however, which I would like to ask. Why are two different steam pressures carried on the boiler plant of the medical college? Would it not have simplified matters to have carried high pressure on all the boilers, reducing it by a pressure regulator for heating?

I would like to know why the steam engines driving the dynamos at the new site were not all made compound—or compound condensing with cooling towers—particularly the large ones, which are to be operated all day long. Would not this have been much more economical in the use of fuel, besides affording an example of good modern practice?

Another question is as to whether the 220-volt system of electrical distribution was considered. A rough computation shows that it would have made a saving of fully $10,000 in copper. In addition, the pressure would have been more uniform throughout the system, and the higher voltage is better adapted for power service, which in this plant is an important feature. Was it on account of the inefficiency of the present 220-volt incandescent lamp and the objections to the 220-volt arc lamp? If the latter, two 110-volt lamps could have been operated in series with entire satisfaction. Another excellent plan would have been to have employed the three-wire 110-volt system.

Another question is as to elevators, which are all electric, except the three in University Hall, which are hydraulic. What special reason was there to justify the increased complication and expense of hydraulic elevators for this particular building?

*Mr. Edward A. Darling.**—In reply to Mr. Bryan's inquiries, first, as to why two steam pressures were used at the medical school, I would say that the five horizontal tubular boilers in the original plant were designed for an extreme working pressure of 60 pounds to the inch, and this was the pressure used for power purposes, as well as for heating. When the plant

* Author's closure, under the Rules.

was enlarged and the use of power for various purposes extended, we decided to secure the economy which attends the use of high-pressure steam and use the old boilers at low pressure for heating purposes only. Undoubtedly it would simplify matters to carry but one pressure in the boiler room and use pressure regulators when needed, and that is the practice we prefer, but in this case the laws which govern the resistance of materials and the regulations of the police department forbade our raising the pressure in the old boilers, and we were unwilling to forego the benefit to be derived from high pressure in the new boilers.

2. The reason why the Corliss en ines at the new site were not compounded was that these engines are run mainly during the winter months, when every pound of exhaust steam is used for heating, in addition to the live steam which is used. Under such circumstances there would be no economy in compounding, but rather a waste in added friction.

By the same token the cooling towers would work our undoing.

3. As to the system of wiring and the voltage used. At the time this installation was laid down the incandescent lamp in vogue required from 110 to 120 volts (at this writing it is still in vogue). As the incandescent lamp was the principal unit to look out for, and we were dealing with accomplished facts rather than future possibilities, the low voltage was adopted. In view of the comparatively short runs, and the added complication in detail which the three-wire system entails, it was thought best to stick to the simple two-wire system.

We have had no occasion to change our views on this point up to date. If we do we can raise the tension.

4. As to elevators. Our reason for using hydraulic elevators in the University Hall was two-fold: first, as the elevators are practically in the same room with the boilers, it did not seem at the time reasonable to us to transmit a load through the generators which could be put on the boilers at first hand, and, second, the ash hoisting is done in the early morning before the large generators are started.

I have no hesitation in saying to Mr. Bryan that, all things considered, in ordinary practice I prefer the electric elevator and advocate its use.

Mr. O. C. Woolson.—I simply want to ask one question of Mr. Darling. I do not think the Society fully understands whether

his 10 per cent. was in favor of the hard coal or the soft coal, and I think it is a very important question.

Mr. Darling.—The difference in economy is in favor of the soft coal. That is my finding—that you save 10 per cent. on the soft coal over the anthracite; but I considered it better not to try to save it.

DCCCXVI.*

EXPERIENCES WITH DEEP-WELL PUMP RODS.

BY G. W. BISSELL, AMES, IOWA.

(Member of the Society.)

THE experiences herein described have been had in connection with the deep well which constitutes the principal water supply of the Iowa State College The well has a total depth of 2,215 feet. Its diameter varies from 12 inches at the top to 5 inches at the bottom, as follows:

From the surface to a depth of			120	feet,	cased	with	12-inch	steam pipe
"	this point	"	"	420	"	"	10-inch	"
"	"	"	"	1,068	"	"	8-inch	"
"	"	"	"	1,430	"	"	6⅝-inch	casing
"	"	"	"	1,935	"	"	5⅛-inch	"
"	"	"	"	2,215	"	not cased, 5 inches in diameter		

The normal level of the water in the well is about 25 feet below the surface of the ground. The level is lowered rapidly by pumping to a definite depth, which is dependent upon the amount of water removed. As ordinarily run the pump removes about 8,000 gallons per hour, and the level of the water is held at about 220 feet below the surface. The cylinder containing the pistons is hung at a depth of 300 feet, being screwed to a 7-inch drop-pipe suspended to the well-head. The cylinder is $5\frac{3}{4}$ inches internal diameter, and of a length suitable for the movement of two pistons each of 24-inch stroke. These pistons are single acting, have conical double-seated metal valves, and are packed with four cup leathers each. Each piston is attached to a string of rods, that for the lower piston working within that for the upper.

In addition to lifting the water to the surface these pistons deliver the water into an elevated tank in which the maximum height of the water level is 150 feet above the surface. Thus

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

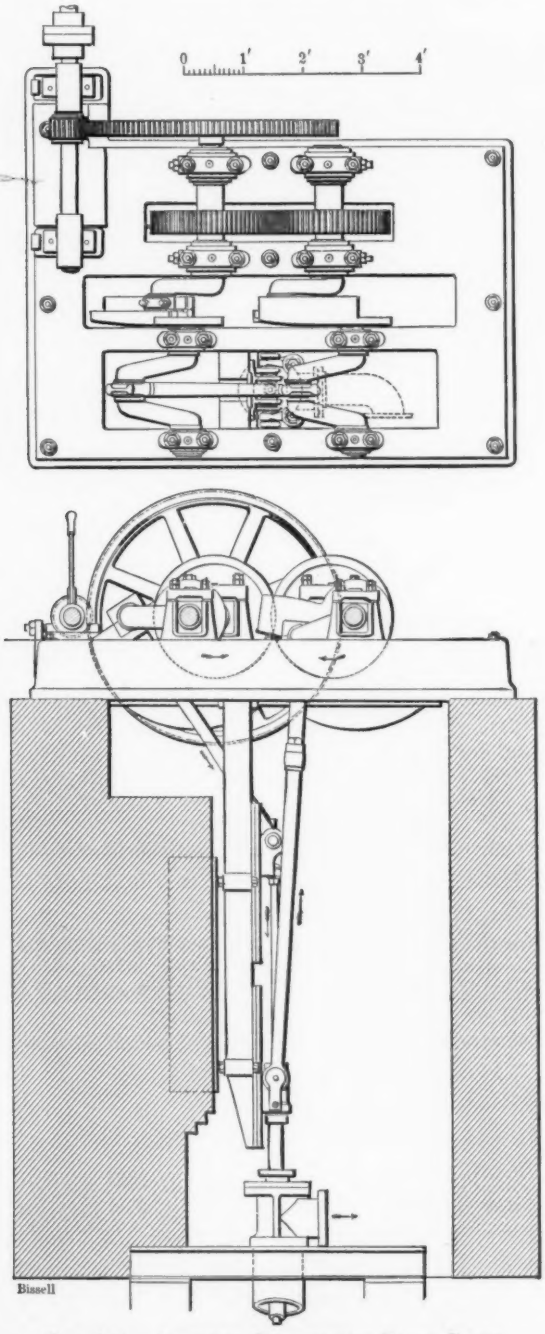


FIG. 253.—JOHNSON CONTINUOUS-FLOW PUMP.

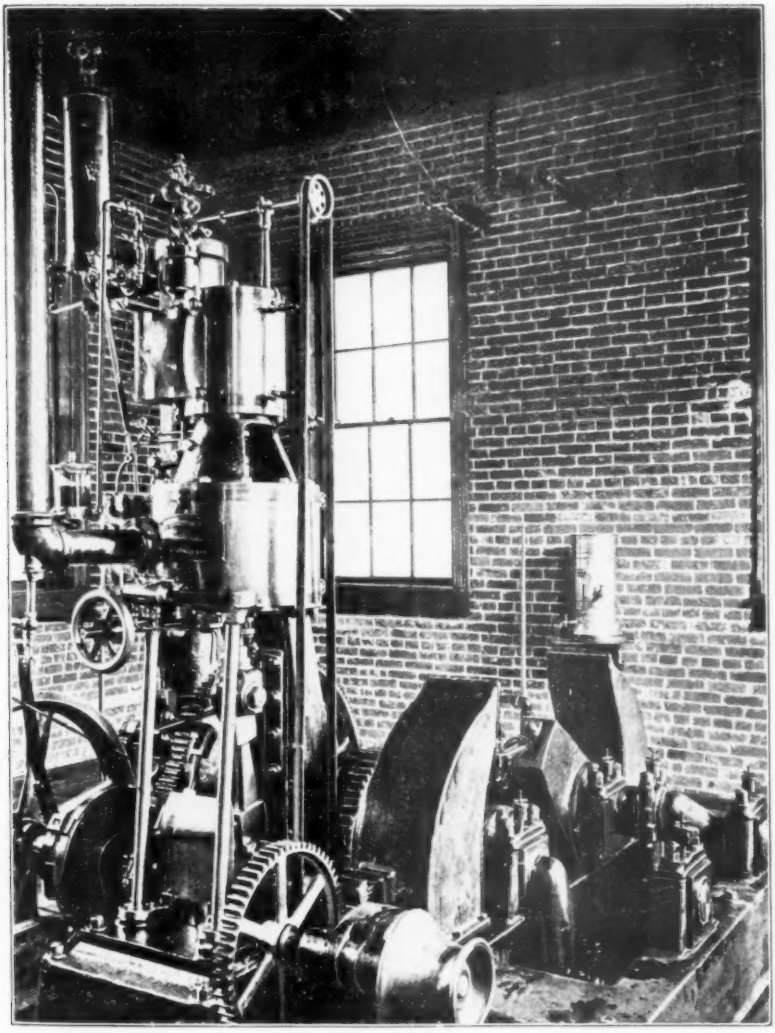


FIG. 254.

when pumping 8,000 gallons per hour the pistons lift the water against a total head of 370 feet. In order to move the water without shock under these conditions it is desirable that the working strokes of the pistons not only alternate but overlap. This can be secured by actuating each piston by a quick return motion. The pump at the Iowa State College embodies this idea. It is known as the Johnson Continuous-Flow Pump, and

is the invention and design of Mr. E. E. Johnson, member of this Society.

Fig 253 shows a plan and elevation of the Johnson pump. The quick return is accomplished by a combination of the Whitworth motion and the offset crank. The mechanism also results in

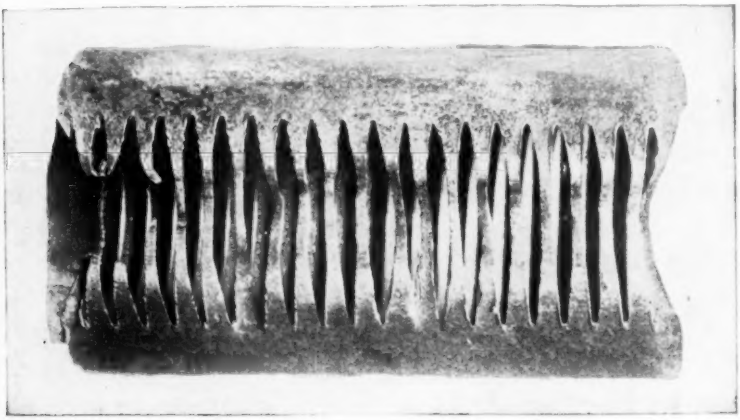


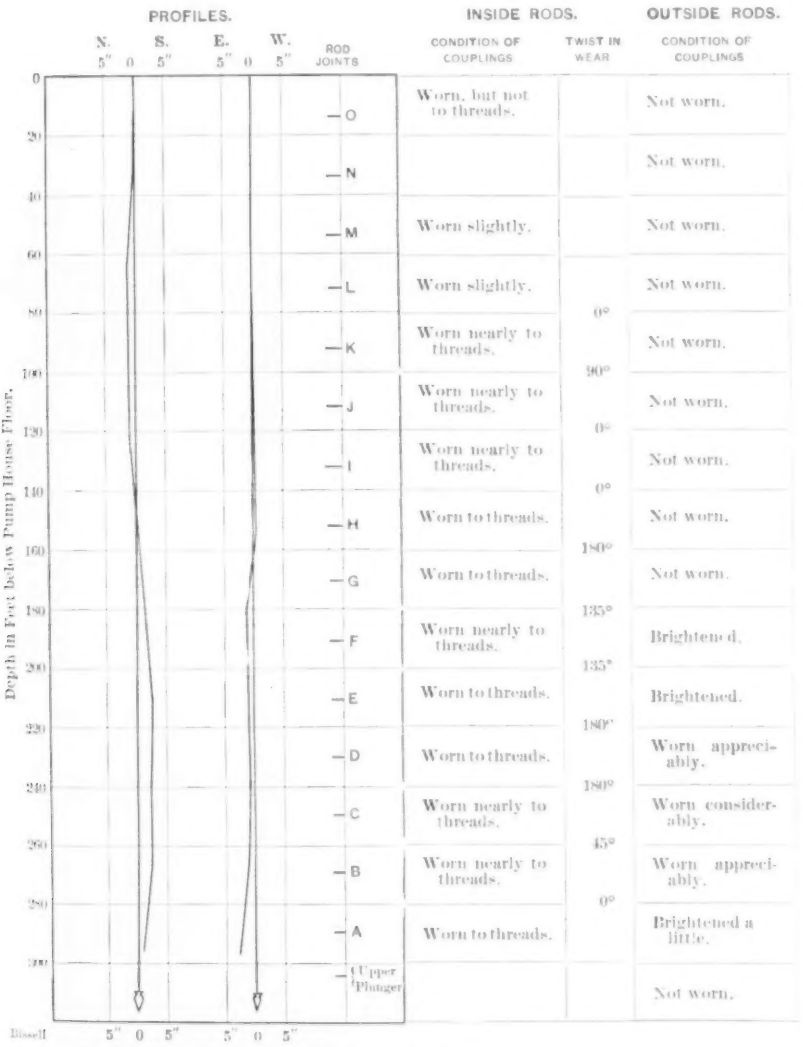
Fig. 255.

having each string of rods and its connecting rod nearly in line during the upward or working stroke. Fig. 254 is a photograph of the pump and engine complete. The engine is steeple compound, 8 and 16 by 8, 200 revolutions per minute, geared to drive the pump at 30 revolutions per minute. At 80 pounds steam pressure and 24 inches of vacuum the duty of this pump is between 40,000,000 and 45,000,000 foot-pounds per 1,000 pounds of dry steam.

Fig. 256.

The inside string of rods originally furnished by the makers of the pump were 1⅛ inches diameter in the body; the ends were upset for a 1¼-inch bolt thread, and had square shoulders for wrenches. The couplings were extra heavy and tapped parallel so that the rods would come together. The outside string of rods

consists of 2-inch pipe, ends upset and connected by very heavy couplings. These rods are made up with pipe tools.



Horizontal scale is magnified twenty four times.
Well casing is 12 inches to 120 feet below pump house floor and 10 inches rest of way.
Drop-pipe is 7 inches.

FIG. 257.

The pump was installed in the fall of 1897. On January 16, 1898, the inside rods parted. Examination showed that many of the inside rods were badly worn at the joints, and that one coup-

ling had been so far reduced as to have parted under the working strain. Fig. 255 shows the appearance of many of the couplings. Fig. 256 shows the wear on the square of the rod near the coupling. Some couplings on the outside rods also showed considerable wear. This fact was taken to indicate that the drop-pipe might be crooked, due to lack of straightness in itself or in the casing of the well. Accordingly the profiles, in two planes, of the centre line of the drop-pipe were determined by means of a wire

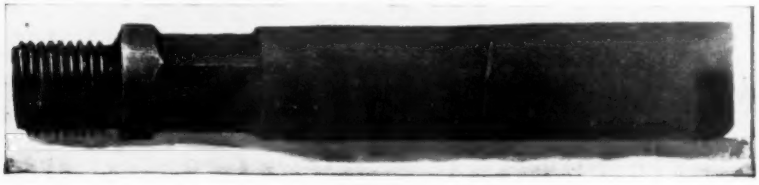


Fig. 258.

cage lowered by a steel wire into the drop-pipe from a fixed point vertically over the centre of the top of the drop-pipe. Deviations of the wire from this centre as the cage descended gave data for plotting the profiles. These are shown in Fig 257, together with a statement of the condition of both strings of rods at each joint thereof. "Twist in Wear" refers to the relation of the wear on the opposite ends of the same rod. Certain coincidences in the

Fig. 259.

wear of the outside rods and the profiles of the drop-pipe led to the placing at the points *B*, *D*, and *F* of brass winged guides, loose on the outside rods, and having broad shoes bearing loosely on the inside of the drop-pipe. Very little wear of the outside rod couplings or of the guides themselves has since been observed. But, as will appear later, the drop-pipe has been removed, and a chance thereby given to rectify or perhaps increase the crookedness of the drop-pipe, provided the same was not due to crookedness of the casing of the well.

The first break and wear of the inside rods were repaired by putting in new couplings. At the same time new inside rods were ordered, but the old ones parted again before the new ones arrived. The second failure occurred February 26, 1898. The new rods were put in place soon after. They differed in design from the old ones in being made of extra heavy 1-inch pipe with ends welded on and provided with couplings smaller than the body of

FIG. 260. FIG. 261. FIG. 262.

the rods. An end of one of the new rods is shown in Fig. 258. On May 10, 1898, the rods were removed to free one of the pistons of a knot of rope which had lodged therein. At this time serious wear was noticed in the inside rods, and its character was such as to lead to the conclusion that the wear was due to the rods not being in line at the joints. Simple tests confirmed this conclusion and the rods were very carefully straightened before replacing in the well. No further trouble on account of the rods was ex-

FIG. 263. FIG. 264. FIG. 265.

perienced until about the middle of October, when removal of the rods was necessary and brought to light one case of serious wear, due to a crooked joint. Fig. 259 shows the end of one of the two rods which were joined at this place. The pipe forming the body of the rod had been worn through. Figs. 260, 261, and 262 show sections of same rod at distances of 12, 24, and 48 inches respectively from the joint, and Fig. 263 shows the coupling and a portion of the rod.

Figs. 264 and 265 show the accompanying wear on the inside of an outside rod. On March 6, 1899, the rods were removed for inspection and found to be in good condition.

An interesting experience with the drop-pipe may be recorded here. During the months of May, June, and July the number of

FIG. 266.

hours of pumping required to keep up with the demand was noticed to increase day by day, the working level of the water rose gradually, and the water became highly charged with air. A tank measurement of the displacement of the pump showed an enormous slip. A leak in the drop-pipe was suspected. Accordingly, early in August the rods were pulled. No defects of

rods or pistons were noted. A test of the drop-pipe with its lower end plugged showed a considerable leakage. The casing was taken out, and at a depth of about 200 feet was found a hole worn by erosion of the water between a coupling and the pipe to which it was attached. Figs. 266 and 267 give a good idea of

FIG. 267.

the size and location of the hole. Evidently a small original defect in the thread of the pipe or coupling had been enlarged by erosion to the dimensions shown. At this point the pressure of the water was at least 150 pounds per square inch. The escape of the water into the space between the drop-pipe and the casing would account for the rise of the working level and the

aëration of the water. No flaw of manufacture could be detected in the material of the pipe.

The thanks of the writer are due to his colleague, Prof. A. Marston, for the profiles of the drop-pipe and for other assistance in the preparation of this paper.

DISCUSSION.

Prof. F. R. Hutton.—It is perhaps because I do not understand the exact mechanical detail of the Johnson pump that I rise to ask the hydraulic engineers of the Society their opinion upon a quick-return mechanism for actuating a pump rod in a deep well. The theory, I take it, is that since the pump is not lifting on the descending stroke, therefore the rod could descend more quickly and by the exertion of a less leverage than upon the working or up stroke. It would seem to me, however, that the volume displaced by the pump rods would force a considerable lifting of water even upon the idle stroke. I have been brought up to believe that there is a considerable waste of energy in unnecessarily accelerating the mass of water which is being pumped, and that water objects to this acceleration with a determination which will put considerable strain upon long rods when it is attempted. May it not be the case that a considerable stretch upon the pump rod results when the quick-return motion occurs, and that this stretch, concurring with abrasive wear during the pushing stroke, has helped the rapid wearing of the rod at points where this stretching was concentrated?

Mr. C. V. Kerr.—I do not get up because I consider myself one of the hydraulic experts; but I think I can throw some light on Professor Hutton's query. I have had something to do with one of the Johnson deep-well pumping stations at Riverside, near Chicago, and we found in that case very much the same wearing of the knuckle-joints in the inside rod that is spoken of here. It seemed to me, however, that it was largely due to the effort on the part of the contractors at that time to get the pumping machinery up to the required speed of thirty-seven revolutions per minute, and that the consequent high speed of the pump caused the rods to buckle; that is, that there was a certain speed at which the effect of gravity would pull the rods down and keep them straight, and if you reached a higher speed than that, then this buckling action took place on account of the effort of the machinery above to

push the plunger down. Now, since last October or November, at which time the higher speeds were run, the pumping machinery has been run at not over two-thirds of its rated capacity. Those pump rods have been taken out since and have been found in good condition, the knuckles not being worn at all. In regard to the other point raised by Professor Hutton, I think that possibly he misunderstands Mr. Johnson's purpose in using the quick-return motion. It is to give a slow movement on the up stroke when the plungers are doing work, and a quick motion on the down stroke when the plunger is simply passing through the water. The idle plunger takes hold of the stream of water just before the working one lets go, and the consequence is, or is supposed to be, that the stream maintains a steady velocity. It is not the purpose of the quick return to accelerate that stream; it is to give the stream a steady motion by giving the working plunger itself a steady motion.

Mr. Francis H. Boyer.—About fifteen years ago I was brought in contact with quite a number of deep wells. We tried the method of using iron pipe for pumping rods. The wear on the casing or the tubing forming the well was such that within a short time the casing would be worn through and destroyed. In the paper that has just been read I recognize an action to which I should attribute materially the wearing of these metallic pumping rods. We have substituted and are using to-day a wooden rod, so made that the joints forming the connections are smaller than the body of the rod, and when it comes in contact with the walls of the pipe of the pump chamber it is wood against iron, and the result is that we have a destruction of wood which we can get out of the well with very little expense, and put in another piece, with a carpenter's work of one or two hours, and save our pump casings. When we were still working with the iron rods we put in what we termed at that time spiders or guides made of composition, soft brass or metal, and let them wear up and down to keep the iron rods away from the casing. These guides, however, would soon be destroyed; while with the wood, if the rod buckled or pushed over to one side, which it does either with iron or wood, little harm would be done. When you think for a moment, even if you have got a single-acting lift pump you are getting a double action by the displacement when the piston and rod descend and force the water up. With a single valve you get the water discharged on both strokes. The down stroke equals the displacement of the piston with its valve going

down. I do not think it is practical to use rods made of pipe, and that is the result of years of experience.

The President.—It is many years ago that I had my experience with deep-well pumping in the Biddy Moriarty oil well, which I bored with my own hands and pumped through 1,700 feet. I used wooden rods altogether, and would not think of trying an iron rod.

*Prof. G. W. Bissell.**—The writer, in view of his own experience, agrees with the idea that wooden rods are better than iron ones, but does not see how they could be applied to replace the outside rods in the college pump. Nor does it appear advisable to revert to the use of one plunger only; under the conditions the Johnson pump gives a continuous flow, and this is very desirable.

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* Author's closure, under the Rules.

DCCCXVII.*

PIPE FLANGES AND THEIR BOLTS.

BY A. F. NAGLE, CHICAGO, ILL.

(Member of the Society.)

THERE is probably no part of a steam plant which presents so untidy, so incongruous, an appearance as the exposed pipe flange joint. The engine itself is handsomely finished, and all its steam joints are steam-tight and covered in a substantial manner with non-conducting material and sheet steel. But we do not get away from this finished and polished machine before we are offended by the sight of rough, uncovered, and hot pipe flanges. Why is this unsightly practice continued? Why should not a simple little round pipe joint be made as steam-tight as any part of the engine?

The practical engineman will tell you that all these steam-pipe joints are liable to leak or blow out at any time, and hence he wants to have easy and quick access to them, which he would not have if they were covered. Here is a confession of a weak point in our steam-engineering practice which should not be allowed to go on forever. I have given this subject considerable study of late years, and some of my conclusions I shall be glad to lay before this Society.

It can scarcely be imagined that it is difficult to make a single flange joint steam-tight when undisturbed by an extraneous force. The fact, however, is that in a modern large steam plant the elongations due to heat are irresistible, and provision must be made to take care of them. This is not always easy to do within the limited space at command, but it may be set down as a guiding principle that wherever possible it is better to let these expanding forces come upon torsional resistances rather than in the form of transverse stress upon the pipe and its flanges. Flexure produces a strain upon the bolted joint which

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

it is not able to stand to any appreciable degree without opening up the joint, but torsion is easily resisted by a bolted joint to any degree even to the point of twisting the pipe to destruction, should it not possess sufficient flexibility to give the required amount of angular motion.

Let us now analyze the forces at work upon a flange joint, its bolts and gasket, for the standard high-pressure flanges (250 pounds per square inch).

The first problem that presents itself is the available power of bolts to make tight joints. In our *Transactions*, vol. xii., page 781, Mr. McBride favors us with a practical test of the lifting power of a common rough screw bolt. These experiments, together with the discussion which followed his paper, clearly indicate that only 10 per cent. of the theoretical power of a bolt may be taken as its effective or actual binding power.

These experiments also show that 200 pounds may be taken as the maximum pull one man can exert for a moment in tightening up a nut.

With these data before us I have constructed Table I., which gives the available power of different size bolts with various lengths of wrenches, and also what length of wrench would be required to obtain the safe limit of stress of metal in a bolt—say 10,000 pounds per square inch.

It will be seen by column 6 that about 10,000 pounds effective binding force is available with one man and the ordinary length of wrenches in use.

Mr. McBride gives it about 7,500 pounds, but I have assumed longer wrenches than were used by him.

It will be seen that a $\frac{3}{4}$-inch bolt should be strained to only 3,020 pounds when its fibre stress amounts to 10,000 pounds per square inch. This is fully as much as it is safe to put on it, for a non-calculable allowance must be made for the weakening effect of the sharp thread ending in a rough and eccentric manner to the body of the bolt, and its head and nut not having a full and fair bearing. These causes make the bolt much weaker than its calculated strength, and 7,000 pounds per square inch would probably be as great a stress as it is safe to put upon the bolt metal.

Column 9 shows what length of wrench would be required, allowing $1\frac{1}{2}$ inches from end to centre of hand, and exerting a pull of 200 pounds, in order to strain the fibre in bolt to

10,000 pounds per square inch. Two hundred pounds is probably more than men will pull on a wrench, still they can do it by bracing their feet.

Having established a pressure obtainable by bolts let us now turn to Table II. and see what the result is when applied to flange joints. Columns 1, 2, 3, and 4 are self-explanatory, except to add that these dimensions are those given by the Chapman Valve Company, and all calculations following are based upon 250 pounds pressure per square inch.

Column 5 gives the pressure per square inch upon the gasket and is found by subtracting from the power of the bolts, found in Table I., column 8, the steam pressure on the area of the pipe, and dividing the result by the net bearing area of the full flange.

For example: A 6-inch pipe flange has 10 $\frac{7}{8}$-inch bolts whose combined pressure = 42,000 pounds.

Pressure in 6-inch pipe at 250 pounds per square inch = 7,075 pounds.

Net total pressure on gasket = 42,000 − 7,075 = 34,925 pounds.

Net area of flange = 105 square inches.

Pressure on gasket per square inch = 34,925 ÷ 105 = 336 pounds.

It will be observed that the pressure upon the gasket is small, going as low as 168 pounds per square inch on a 16-inch pipe, and the highest is 347 pounds per square inch on a 3½-inch pipe. With 250 pounds steam pressure these pressures upon the gasket are small, and it is not surprising that when a slight transverse stress is produced by expanding pipes the joint is opened on one side and leaks begin.

We naturally turn to the need of a tongue and groove to reduce the bearing area of the gasket.

The groove also has the merit of protecting the gasket from misplacement, or being forced out in part by the pressure of the bolts; the gasket is also protected from being blown out if slightly relieved on one side. The disadvantage is that the gasket is not renewable without opening up the space between the pipes several inches, which is not always possible. But I am pleading for a system of piping and joints which will not blow out or need renewal.

If we turn to column 6 we shall find the width of tongue used by the Chapman and Crane companies. One-half ($\frac{1}{2}$) inch is too

narrow for practical safe handling of gaskets. Column 7, however, shows the effective pressure on these narrow gaskets. In the smaller sizes the pressure runs up to 2,500 pounds per square inch, and in the larger sizes it goes as low as 600 pounds per square inch.

It would appear at once that some more nearly uniform pressure upon the gaskets would be desirable, and I have therefore assumed that 1 inch width for all sizes of pipes would be a very good standard to use. Still retaining the Chapman bolt table, columns 8, 9, and 10 will show the result of using this width of tongue. It will now be seen that the pressure per square inch of gasket varies but little from 1,000 pounds, although a few extreme cases go from 755 pounds to 1,360 pounds.

We could use said bolt table, but change the width of tongues to one inch, and yet get very much better practical results than with existing proportions, but to get still better and more uniform results as indicated in column 17 I have revised the number and sizes of bolts slightly. Columns 11 and 12 give the Chapman bolt table, columns 13 and 14 the Crane, and 15 and 16 the one I recommend. It will be observed that I have not created any new sizes, but have used in part from each of the above-mentioned tables.

The Crane table evidently has aimed to make the number of bolts divisible by 4, so as to permit turning valves quartering after being drilled; whether this is of sufficient practical importance to offset the better holding of joints I am not prepared to say, but we see now what is desirable for bolts and gaskets, and we can come as near to its fulfillment as circumstances will admit of.

Up to and including 9-inch pipe I have not changed the Chapman bolts, but for 10-inch and 12-inch I have adopted the Crane table, namely 12 and 16 bolts 1 inch in diameter respectively, instead of 15 and 18 seven-eighths ($\frac{7}{8}$) bolts, not simply because they give somewhat greater pressure on the gasket, but for reasons already pointed out, namely, that the $\frac{7}{8}$-inch bolt is below the safe minimum size bolt pointed out in Table I. I will also add that my personal practice has banished the $\frac{7}{8}$-inch bolt from use on these sizes of pipes years ago, because I have found it would not stand the strain put upon it in practical use.

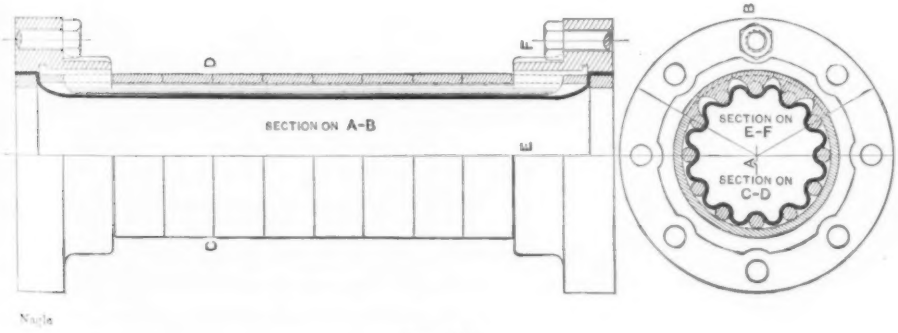
On the 16-inch pipe the Crane 20–1-inch bolts give better results than the Chapman 18–1-inch, but I would recommend

this number and size be increased to $1\frac{1}{8}$ inch. On the 15-inch pipe, a rather odd size, I would also make 20–$1\frac{1}{8}$-inch bolts instead of 18–1-inch. In all sizes above 16-inch these two manufacturers agree, but I recommend that the 22-inch pipe flange-bolts be increased from $1\frac{1}{8}$ to $1\frac{1}{4}$ inch, the number (24) remaining the same.

It is striking how much more uniform pressures upon the gaskets are produced by a standard width of tongue of one inch than by the present practice, and it seems to me that the adoption of such a standard would be very desirable.

In column 9 are given the inside diameters of tongues if my recommendations were adopted.

The tongues and grooves of the Crane Company are of precisely the same diameters as those of the Chapman Company.



Nagle

FIG. 268.

The projection of tongues in Chapman tables varies from $\frac{3}{16}$ inch to $\frac{7}{16}$ inch; the depth of the grooves is not given. In the Crane tables the projections of tongues vary only from $\frac{3}{16}$ inch to $\frac{1}{4}$ inch, and the depth of grooves is always $\frac{1}{16}$ inch less than the tongues.

I would recommend for adoption in all cases a uniform tongue projection of $\frac{1}{4}$ inch and a depth of groove of $\frac{3}{16}$ inch.

One word about the thickness of flanges. When the tongue and groove construction is used the flange becomes an overhanging beam and it should be made very strong. I would recommend that in all cases the flange be thickened at the bolt with a hub extending back as far as the hub of the flange. This is a construction already in use by some manufacturers in some of their flanges.

In order to carry out more completely the writer's idea of relieving flange joints of all transverse stress, or flexure, and substituting, so far as possible, torsional strains therefor, I have designed a pipe construction which admits of torsional movement to a large degree without straining either the material of the tube or the flange joint to any appreciable extent. It is shown in Fig. 268 and consists of what might be termed stave construction instead of solid pipe. These staves are steel rods

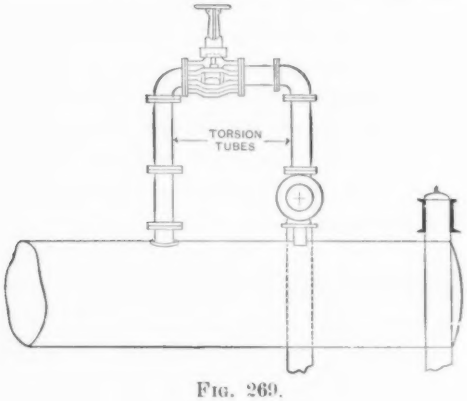


FIG. 269.

securely fastened to the flanges. Exterior to these staves are strong iron or steel enveloping rings. Interiorly is a thin, longitudinally corrugated copper tube also securely fastened to the end flanges. We now have a pipe which is strong in every direction but one, namely, that to resist torsion. With these flexible tubes put in in the proper manner all elongations of steam pipes should be easily taken care of and without straining any joint to a dangerous degree. See Fig. 269.

I do not intend to patent this form of pipe, and hence it can be made by any one who chooses to do so.

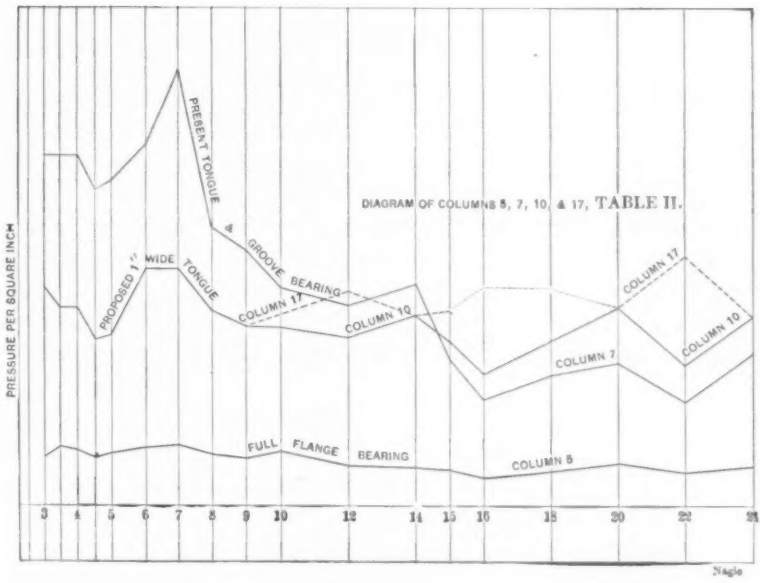


TABLE I.

Diameter of Bolt.	T= Threads per Inch.	Length of Wrench	C= Effective Circumferences.	P= Theoretical Pull of Screw.	Effective Pull.	Area of Bottom of Thread.	Pull of Bolt of Metal per Sq. In.	Length of Wrench.
Inches.		Inches.	Inches.	Pounds.	Pounds.	Inches.	Pounds.	Inches.
$\frac{3}{4}$	10	10	53	106,000	10,600	.302	3,020	4
$\frac{7}{8}$	9	10	53	95,400	9,540	.420	4,200	$5\frac{1}{2}$
1	8	12	66	105,600	10,560	.550	5,500	7
$1\frac{1}{8}$	7	12	66	92,400	9,240	.694	6,940	$9\frac{1}{2}$
$1\frac{1}{4}$	7	12	66	92,400	9,240	.893	8,930	12
$1\frac{3}{8}$	6	16	91	109,200	10,920	1.06	10,600	16
$1\frac{1}{2}$	6	16	91	109,200	10,920	1.30	13,000	18
$1\frac{5}{8}$	$5\frac{1}{2}$	16	91	100,100	10,010	1.52	15,200	24
$1\frac{3}{4}$	5	18	103	103,000	10,300	1.75	17,500	30
$1\frac{7}{8}$	5	18	103	103,000	10,300	2.05	20,500	36
2	$4\frac{1}{2}$	18	103	92,700	9,270	2.30	23,000	42
1	2	3	4	5	6	7	8	9

3. Length of wrench customarily used in shop practice.
4. Application of hand taken at $1\frac{1}{2}$ inches from end of wrench.
5. Pull or force of man taken at 200 pounds, or $P = C \times T \times 200$.
6. Effective Pull = 10 per cent. of last column.
8. When metal is strained to 10,000 pounds per square inch, bolt is strained, or will resist, as per this column.
9. (Length of wrench $- 1\frac{1}{2}$) $\times 3.1416 \times T \times \frac{2000}{10} =$ column 8.

TABLE II.

Diameter of Pipe.	Diameter of Flange.	Bolts.		Net Pressure on Gasket per 1 Sq. In.	Width of Tongue.	Net Pressure on Gasket per 1 Square Inch of Tongue.	Width of Tongue.	Inside Diameter of Tongue.	Net Pressure on Gasket per 1 Square In. of 1 inch Tongue.	Chapman B'lts.		Crane Bolts.		Nagle Recommends.		Net Pressure per sq. in. on Gasket 1 Inch Tongue, Nagle B'lts.
		Number.	Diameter.							Number.	Diameter.	Number.	Diameter.	Number.	Diameter.	
Inches.	Inches.		Inches.	Pounds.	Inch.	Pounds.	Inch.	Inches.	Pounds.		Inches.		Inches.		Inches.	Pounds.
3	9	6	¾	290	½	2,000	1	3⅜	1,254	6	¾	8	⅝	6	¾	1,254
3½	9	7	¾	347	½	2,000	1	4	1,150	7	¾	8	⅝	7	¾	1,150
4	10	8	¾	320	½	2,000	1	4⅝	1,130	8	¾	8	⅝	8	¾	1,130
4½	10½	8	⅝	288	½	1,800	1	5¼	950	8	¾	8	⅝	8	¾	950
5	11	9	¾	300	½	1,857	1	5⅝	975	9	¾	8	⅝	9	¾	975
6	13	10	¾	336	½	2,057	1	6¾	1,360	10	¾	8	⅝	10	¾	1,360
7	14	12	¾	350	½	2,500	1	7⅝	1,360	12	¾	12	⅝	12	¾	1,360
8	15	12	¾	300	½	1,600	1	8⅝	1,130	12	¾	12	⅝	12	¾	1,130
9	16	13	¾	283	½	1,474	1	9⅝	1,030	13	¾	12	⅝	13	¾	1,030
10	17½	15	¾	267	½	1,250	1	11	1,030	15	¾	12	1	12	1	1,110
12	20	18	¾	236	½	1,160	1	13	963	18	¾	16	1	16	1	1,240
14	23	18	1	231	½	1,270	1	15	1,095	18	1	16	1	18	1	1,100
15	23½	18	1	213	½	830	1	16	925	18	1			20	1	1,130
16	25	18	1	168	½	618	1	17	755	18	1	20	1	20	1¼	1,200
18	27½	24	1	200	1	754	1	19	972	24	1	24	1	24	1¼	1,228
20	29	24	1¼	257	1	930	1	21	1,160	24	1⅛	24	1¼	24	1¼	1,160
22	31	24	1⅛	191	1	600	1	23	827	24	1¼	24	1¼	24	1¼	1,450
24	34	24	1¼	222	1	888	1	25	1,100	24	1¼	24	1¼	24	1¼	1,100
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17

DISCUSSION.

Mr. F. A. Halsey.—The sketch, Fig. 270, shows a form of joint with which I became acquainted some years ago, and which was so satisfactory under trying conditions that it deserves to be more widely known. It was invented by Captain Rapieff for use with the dynamite gun—that is, the gun commonly known as the Zalinski gun. The joint was applied to the gun itself, which is in sections, and throughout the pipe fittings. The receivers are sectional, with numerous connections, and positive tightness and reliability are essential. The firing reservoir, the gun, and the

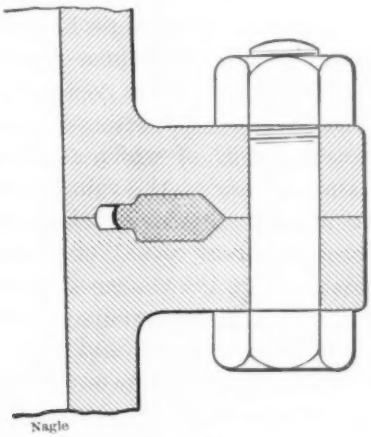


FIG. 270.

intermediate connections operate under a pressure of 1,000 pounds per square inch, while the compressor, the storage reservoir, and their connections operate under 2,000 pounds.

The sketch shows a section of the joint. A groove of the section shown is turned in the face of each flange, a surplus or "overflow" space being provided at the left. In the space formed by these grooves a strip of round soft rubber is inserted, the cross-sections of groove and rubber being so proportioned to each other that when the joint is closed the rubber will occupy part of the overflow space. The rubber used is not a continuous ring and is cut from a long strip with a scarf joint where the ends meet.

It will be seen that the pressure coming down the joint drives the rubber into the angle at the right, and that leakage is prac-

tically impossible. The grip of the bolts is not relied upon to retain the rubber by friction, as with the ordinary gasket. The joint is metal and metal, and the pipe sections may be gotten out to defined lengths—a feature which is occasionally useful. In the case of the gun, an additional feature of great value is that the tightening of the bolts does not, as with gaskets, tend to throw the sections out of line.

As stated, this joint was used in a great number of cases, and I was assured that failure with it was unknown—positive tightness being expected and obtained as a matter of course.

Mr. Reginald Pelham Bolton.—While it is perfectly clear, as Mr. Nagle says, that some improvement is necessary in flange pipe joints, I do not think his paper altogether goes far enough. He has not given us a comparison, as he might have done, of the flanged joints which he has tabulated, with those joints which were laid down, I believe by this Society, some years ago, as a standard, and also those which were brought into existence by the Master Steamfitters' Association, both of which are largely made use of. He would thus, I am sure, add to the value of the paper. It also appears to me that the difficulty under which we are laboring is this—that there is such a great variety of pressures that no uniformity can be expected in the matter of joints. For instance, in steam work, it is only of recent years that a pressure of 200 or 250 pounds has become common, and now it is necessary to design joints for that pressure; but our difficulties even with pressures as low as 120 pounds are very considerable. The paper is chiefly devoted to dealing with flanged joints in which a recess exists, and there is considerable difficulty always raised in the use of those, by steamfitters, on account of the difficulty of disconnecting them, and there is very much prejudice against their use on that account. It is a considerable difficulty, because that little tongue which has to be gotten out of the groove when you break a joint involves taking down a whole line of pipe to get down one piece. It seems to me that is so serious that something better than the tongue and groove is necessary for general use, if a standard is to be laid down for all pressures. Some such joint as has been described by the last speaker would probably meet the objection which I raised, but that of course is not in common practice. Mr. Nagle's detail of a piece of torsional pipe may possibly be of considerable value in certain locations, but certainly would be an expensive construction, and it appears to me

to be somewhat unnecessary, inasmuch as these torsional strains all arise from expansion, and if the expansion is properly cared for the torsional strain will practically be reduced to nothing. In long lines of pipe expansion-joints are commonly made use of, and the extensions sideways from a long line of pipe are taken off in such a manner as will allow of a known spring in the pipe itself, which, therefore, does not bring a great strain to bear on the flanges. I recently fitted up a line of main steam pipe 300 feet long in which the expansion was quite considerable, and we have not any expansion-joint in at all. There are five spring connections taken out of the top of the line for engines, the end of the line 200 feet away from the boilers, these simply being carried out beyond the engine and brought back again in a U shape and down to the engine. These joints have given no trouble whatever.

Finally, I would like to note that, in those sizes of flanges and the number of bolts which Mr. Nagle recommends, it appears to me that he has not recommended sufficient strength. I have had experience, for instance, with a 12-inch steam pipe at 135 pounds pressure where a flange of 20 inches in diameter with 18—$\frac{7}{8}$-inch bolts could not be kept tight, but as soon as those flanges were replaced by heavier ones having 18—1-inch bolts in there was no further trouble experienced. As the writer says, the question of the thickness of the flange has more to do with it than anything else. Most of the troubles I have found in flange-pipe work have arisen from the distortion of the flange due to securing the pipe into the flange, as is commonly done, screwing it in tight, or expanding it into place. Either will be very apt to throw the flange out of shape. It seems to me that proper practice would call for the flange to be turned up in a lathe after the pipe had been made tight into it. It is probable that what we shall arrive at in no great length of time will be a more general use of pipe formed with the flange in one piece, such as the cast-steel piping which is being successfully used in Providence and elsewhere.

Mr. James McBride.—At a previous meeting of this Society I had something to say on the pipe joint. I can only reiterate what I said then. I think the great difficulty with the commercial flange as we find it sold by dealers is that it is not heavy enough; it has not got stock enough in it. As we say in Western woods phrase, it hasn't "got the backbone in it." We have had a great deal of trouble with steam-pipe flanges and water-pipe flanges, principally with steam pipes. I eliminated, I might say, 90 per

cent. of all my trouble by making my flanges three times as heavy and putting plenty of bolts in them and making the bolts large. Make your flange of anything you are of a mind to, but make it strong; give it strength, so that all the strain you can put on the bolts will not distort it. I agree perfectly with the gentleman who spoke last, in regard to making a flange recessed. If anybody has ever had any experience in taking apart the joints in a vacuum pan where they are made perfectly rigid and there is no spring whatever, I think he will agree that the plain surface flange is the proper one, and there are lots of other places where short steam pipes are stiff and rigid and where it would be practically impossible to take them apart in the time in which repairs have got to be made. I think that a great deal of trouble can be eliminated by simply making the flanges very heavy, putting in plenty of bolts, and making the bolts large.

Mr. Wm. Kent.—I think that the fact that Mr. McBride has not had any trouble with his steam pipes has not been due entirely to his making his flanges good and strong, but also to the fact that he did some good designing on his pipe hanging beside: that is, he did not introduce these transverse strains, which will make any kind of a flange leak. He has certainly remedied one of the great troubles of the steam-pipe business, which has been to make flanges too weak. But a fully equal trouble is imperfect design, by reason of which transverse stresses were brought into the pipe. I have had occasion to look into the steam-pipe question a good deal in the last two or three years, and I think it is the most dangerous and unsatisfactory portion of the whole steam plant. It is high time that our whole practice in steam piping was revolutionized. I think one of the things to do is to get rid of the cast-iron flanges altogether and cast-iron short elbows and come to forged steel flanges and bent steel elbows.

Mr. H. H. Suplee.—In this connection the whole discussion has turned toward a line which has been worked out very carefully in Germany by Professor Bach, and his paper recently presented before the Society of German Engineers contains much valuable information on this subject. He experimented with pipes and fittings and the connections for flange joints, using small contact rods touching the various portions of the fittings and provided with fine micrometers. By putting on a heavy pressure and then releasing it, the movement of these various rods showed the distortions which had taken place in the fittings. In nearly every case it was found

that there was a distortion produced by the pressure, which tended to throw the flanges out of truth, the strain passing beyond the elastic limit, so that flanges did not come back to the original position. The whole result showed that the pipe flanges and fittings proportioned according to the old rules and formulas are too light for the pressures we are now using.

Mr. J. E. Johnson, Jr.—A point which I think has not been brought out in this discussion, although deserving of the most careful consideration, is the effect of the width of the joint face proper upon the possibility of making a joint tight with a given number of bolts.

It is necessary to remember that the surfaces in contact must be drawn together at *all* points with a pressure at least equal to that of the fluid to be held, or else there is going to be a leak.

With gaskets which are comparatively soft, even when made of metal, this is generally not difficult, as the pressure per unit of surface produced by any reasonable proportions of joint face and bolt strength will cause a flow which will close up all fine lines or grooves across the face of the joint; but when the joint is made metal to metal it is necessary to produce sufficient pressure to obliterate all such fine, almost microscopic, imperfections in the joint faces by forcing them together with such a pressure as to cause the metal of the opposite face (in each case) to flow into them.

The pressure per unit of area is simply the total pressure divided by the area in contact, and therefore it will require in any given case more than five times the bolt pressure with a joint face one inch wide to produce tightness under a given pressure as it will with a joint face three-sixteenths of an inch wide.

To the speaker's knowledge joints are being made to-day on the lathe and boring mill without any subsequent grinding of any kind, and without any especial tooling or pains whatever, which are tight from the start, by simply having the joint face only about three-sixteenths of an inch wide, and making the flange stiff, so that the bolt pressure comes on the joint and not at the outer edge of the flange.

This latter is a very important point also, and one to which too little attention is paid.

The enormously heavy flanges which several of the speakers have advocated here to-night would not be necessary if the joint face was at the inside edge of, and raised above the surface

of the rest of, the flange, so that the flanges could not possibly come into contact except inside the line of bolt holes. Where this is not the case, but the face of the flange is flat clear across, the effect of setting up the bolts is to draw the outer edges of the flanges together by springing them, so that even if a joint is made at the outside edge leakage can take place through the bolt holes. The excessively heavy flanges which have been advocated here to-night are a partial remedy for this trouble, but a much simpler and better one is to so design the joint that the surfaces in contact *must* be inside the circle of bolts, which is most simply accomplished by having a raised strip like a chipping strip as wide as may be thought best at the inner edge of the flange on *all* pipes; then there are no unlike ends to match and no difficulty in getting out any piece of pipe required, without disturbing the rest of the line. When gaskets are to be used, the speaker ventures to think, with all due deference to Mr. Rockwood, that it is much better to have them small enough to go inside the circle of bolts altogether, as the difficulties of handling them in this way are not great, and it accomplishes the double purpose of keeping the joint narrow and keeping it within the circle of the bolts, so that leakage through bolt holes ceases to be a factor. The writer saw the result of following the opposite course, that is, making the gasket the whole width of the flange, well illustrated not long ago, on a 2-foot pipe carrying air under a pressure of about 4 pounds per square inch.

The original joint was made with a hoop of wrought iron of small size (about quarter-inch) wrapped with cotton wicking soaked with red-lead. The pipe stood with this joint for many years, and leaks were never thought of. Finally it was taken down to be re-erected in another place, and to save the trouble of bothering with the red-lead and wicking, and, as it was supposed, to make an extra good job of it, thick gaskets of rubber composition of the best quality, the full width of the flange, were ordered. They were put in, and nearly all the joints leaked. The number of bolts in each flange had finally to be doubled to make them tight.

The explanation was simple—the pipe was steel-riveted with angle-iron flanges. The old joints were tight because they were narrow and the pressure per unit of area was high, because the joint was made entirely inside the bolt holes, and finally because the flanges had plenty of room to spring under the pressure of the

bolts without coming into contact at the outer edge. In the new joint the pressure per unit of area was only a fraction of what it was before and by far the greater part of it was at the outer edge of the flanges, so that, while many of these were tight *there*, the bolt holes, being inside the joint, were free to leak, and did so.

Prof. Forrest R. Jones.—Not long ago I had occasion to investigate the forms of pipe fastenings commonly used for high pressures.

Among these connections I found one which I believe is fairly modern and which the chief engineer of one of our largest traction companies had adopted. It was a rolled-steel flange. The pipe was fitted accurately to the bore of the flange. The latter was heated so as to expand it, and then shrunk on the end of the pipe. The end of the pipe was then either rolled or peened over to fit the rounded corner of the bore of the flange. The end of the pipe was faced off flush with the flange, so that the one was counterbored and the other fitted into it, the packing lying in this counterbore. The object in doing this was to get the packing so placed that it would cover the joint between the pipe and the flange. He said that he had found that this particular form of joint had given perfect satisfaction throughout, while he had had a great deal of leaking with the old flat-faced joint, with packing between the faces. He had not, for his work, adopted at that time what he said he would if he had larger pipes and higher pressures to deal with; that is, he would have threaded the end of the pipe for screwing it into the flange, and would have peened the ends over and faced them off as before. What I want to call particular attention to is this use of the gasket for covering the joints between the flange and the pipe which fits into it, thus avoiding leakage between the flange and the pipe, as well as between the two flanges.

Mr. Francis H. Boyer.—If this meeting could take the position of the old class meeting, with yourself or the secretary as leader, and you should speak over to this one and say "What is your experience?" and to that one "What is your experience?" we might have some grand things. We have got men here who have had a great deal of experience with pipe flanges under very heavy pressures, and one of the best informed would be an ice-machine man, like our friend George Richmond. I don't know why they should not get up and tell what their experiences are. If he will not I will do the best I can for him. In handling work for the lique-

faction of gases we, for instance, have to meet hard conditions, and in my practice at home a week doesn't go by that we don't make flanged joints that we test under hydrostatic pressure of 1,000 pounds to the square inch. We do not use any class of work for ammonia, either with litharge or lead-soldered joints, without testing it to 1,000 pounds to the square inch, and we never have any trouble. We take and put a small recess tongue and groove between the flanges and insert a lead ring, common ordinary lead. Sometimes we have it made free of sulphur, called chemically pure lead. In making these flanges we invariably put in never less than four bolts. Sometimes bolts break; bolts come defective from the factory, and break without any apparent excessive strains. Now then we put in four bolts, so that if one bolt lets go we have got three more to hold. We follow that practice down to half-inch pipe. Where there is a charge of $4,000 or $5,000 worth of ammonia gas under a pressure of from 100 to 215 pounds to the square inch, it is a pretty bad matter if a joint should get loose; if you do not get your men away you have got a big loss for a life to pay. For that reason we put in four bolts to secure in case of an accident. I have known cases where we have had pipe tested as high as 2,000 pounds to the square inch to rupture. When we came to test pipe with air in starting a refrigerating or ice-making machine, and we pumped the air too rapidly, the oil used to lubricate the cylinder walls of the gas compressor, coming in contact with the heat given off by the compressed air, finally reached the explosive point of the combination of air and oil, and a rupture of piping follows, with a terrific report, destroying joints and pipe work which had previously been tested to 1,000 pounds per square inch. Now, in reference to the expansion and contraction of pipes, I do not believe there is a joint made that can withstand it. I have seen in cases of that kind where it stripped the threads. The trouble comes in making this class of work, that, as a rule, they do not put in a sufficient amount of metal to make work of that character. Mr. McBride got it right; he makes his own patterns and sends them to the foundry, and has them made and brought back and puts in about four times the amount of metal which is used by manufacturers, and then you are on the safe side. Then get in bolts enough so that if you have an accident you have more bolts than are necessary to hold it. I am speaking now of handling gases, and that is the way in which we handle them and do it successfully.

Mr. McBride.—I have tried both methods which have been referred to, and I have done it very carefully. Neither gave any satisfaction, because on the lower part of the pipe, if it is a horizontal pipe, the condensed steam will lie in the recess and will finally destroy the joint, will cut out that narrow part of the joint which lies between the ends of the wrought-iron pipe, and then the joint begins to leak. The same thing occurs with the thread. I have tried those methods so thoroughly that I have given up attempting to make joints tight around the collar of the flange by those means. In the last pipes which I put in I adopted that section referred to by Mr. Kent. There is welded on to a steel pipe a steel flange, faced true. I think that is a finality in making a perfect joint.

The President.—When we are putting in steam pipes 16 to 20 inches in diameter, from 25 to 30 or more bolts in the flange are necessary to put the bolts close enough together to make the joints, and not only that—to make the bolts so strong that when a strong man puts a stress on the bolt he is neither going to strip the thread nor break the joint. We had more trouble recently in the war with Spain with leaky joints than with anything else on our ships, and we have come down to the welded flange.

Mr. Kent.—The whole cost of the steam piping of a large flange is but a small portion of the total cost of the plant, and I think we should be willing to spend the cost of getting the steam pipes right.

Mr. Reginald Pelham Bolton.—I would like to call attention to a good joint (Fig. 271). The tube is extended through the flange and turned right over and flattened out and faced, and the joint is made on the end of the pipe itself, and that with a very strong flange makes a very good joint. I have not, however, had personal experience with that.

Mr. Bryan.—You use no gasket?

Mr. Bolton.—With an iron pipe use a copper gasket; use it large enough and extend out beyond the joint so that it centers itself. One of the troubles which I have experienced in this class of piping has been with the workmen, who not only put that strain on the bolts which is due to the use of a long wrench, but I constantly find them using a hammer on the end of the spanner, and therefore they strain these bolts probably over what they are capable of standing. That I am sure has been a cause of failure of some of the joints, for which I and others sorrow to-day.

Mr. Geo. I. Rockwood.—In the Rockwood joint the pipe is intruded into the flange a sufficient distance to allow the end of the pipe to be turned over against the flange while it is hot. Before flanging the pipe it is rolled into the recess *A*, Fig. 271, by an expander. Now, unless that recess is cast with sharp corners, as at *A*, it is impossible to make the pipe seize hold of the flange by the expanding process. It will spring away again and the flange will be loose on the pipe. Mr. Van Stone, superintendent of the Walworth Manufacturing Company in Boston, has made many of these flanged joints, and the sharp-edge feature was suggested and put into practice by him. I prefer to rivet the skirt of the flange to the pipe, as there can then be no question of the security of flange to pipe.

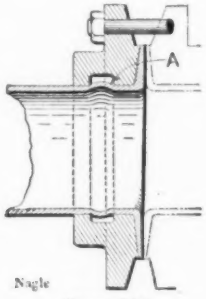


FIG. 271.

The chief advantage of this pipe joint is the fact that steam does not come in contact with the cast-iron flange, as the joint is made between the abutting ends of the pipe, and their edges may be caulked if necessary.

Mr. McBride.—I would like to ask the gentleman if the sketch shows copper or iron pipe?

Mr. Rockwood.—The pipe must be best charcoal-iron pipe or steel pipe. The ordinary commercial pipe will not stand the flanging process without splitting. There is, however, no difficulty in getting a pipe which will stand it. Boiler flues are made of the right kind of stock.

Mr. John T. Hawkins.—I would like to add my endorsement to the remarks of one of the previous speakers to the effect that the proper thing for high steam-pressure piping, so far as flanges are concerned, is to have them made also of rolled steel. Possibly steel castings would be nearly as good. I think the day is coming when such a mechanical abortion as a steel steam pipe with

cast-iron flanges screwed upon it, the pipe being peened or rolled on the inside to prevent tearing the threads off by longitudinal stress, will be abandoned. It has got to be now for stationary work that there is quite a distinct line drawn between what is known as low-pressure and high-pressure pipe, and in the latter case, if you merely specify that you are going to carry a high pressure of steam, and want homogeneous steel piping for the purpose, the contractors will generally put in their specifications "extra heavy fittings," which you will find includes cast-iron flanges. Now, if we screw a cast-iron flange with six or eight bolt holes in it on a steel pipe and then rivet or peen or roll it over on the inside to prevent it being drawn off, we are bringing strains on the cast-iron flange that are not only unknown but unknowable; for it is practically impossible to measure them. For my part I have got to look at that sort of structure as a mechanical abortion, and hope to see the day when rolled-steel or cast-steel flanges will always accompany steel piping for high-pressure steam, unless, indeed, the flanges be formed direct upon the pipe, as I understand is now practised in Europe to some extent.

Mr. Henry C. Meyer, Jr.—Several gentlemen have referred to the excessive cost of the welded steel flange. I have been informed by the National Tube Works Company that the cost of piping with this type of flange is about 25 per cent. greater than with the flange screwed on the end of the pipe in sizes up to 8 inches. From 8 to 16 inches the cost, I am told, is about 10 per cent. greater, while in sizes above that the cost is less.

Mr. McBride.—I can supplement that by saying that the cost of two lengths of 8-inch pipe 16 feet 6 inches long each, with 14-inch diameter flanges, delivered in Williamsburgh, from the National Tube Works, was $83.

Mr. C. P. Higgins.—In regard to the incongruity of using cast-iron fittings on steel pipe, I think the uniform elasticity of the structure would be enhanced by making our fittings of forged wrought steel. And the problem is not as difficult as many might imagine. I have had some experience with wrought-steel fittings, and in the installation of hydraulic work, with pressures of 1,000 to 5,000 pounds, I have found them very efficient indeed. I have equipped steam systems with them and never have been troubled with the failure of one of these fittings, that is, where they are properly made and properly applied. The process of making tees, elbows, and flanges is a very simple one, and one which I hope to

see come into general use, to make the fittings and pipe in keeping, so that we form a structure of wrought-iron or wrought-steel plate uniform throughout and able to withstand the strains that come from contraction and expansion and the movement of pipes, which we get in marine use; and in addition to that, we make our entire structure lighter, easier handled, easier retained in position, and in every way better and more fully in keeping with the best engineering practice.

Mr. A. F. Nagle.—I have but a few words to say, and that is that hundreds of devices have been made to make the joints between the flange and pipe. I think we are free from that trouble since the National Tube Works make a welded flange, I think, of above 5 or 6 inches in diameter. I do not think I shall ever use any more screw flanges above that size. In regard to the price, the National Tube people tell me that for five or six inches the price is about 25 per cent. more than for the ordinary joint, and for about 10 or 12 inches the price is about the same, and when it comes to 24 inches and upward the price is cheaper than the attached flange. With these facts before me I think engineers are through with trying to make joints between flanges and pipes. The other matter, I think, has been sufficiently discussed in the paper, without discussing the matters as to tongues and grooves and size of bolts.

Prof. Forrest R. Jones.—I would like to ask if any such trouble has come to this joint of Mr. Bolton's as Mr. McBride has described, of corrosion by the water gathering in the groove formed at the end of the pipes.

Mr. Bolton.—I think I already answered that by saying that I had not had experience with the joints, but I should imagine that that and all other joints made in the same way would suffer from that danger, although it might be a long time before it took effect.

Mr. McBride.—In my experience with that kind of joints I found that all the joint material was gone between the ends of the wrought-iron pipe on the lower side of the pipe where the condensed steam lay.

*Mr. A. F. Nagle.**—It is perhaps only fair to myself to say that it never occurred to me that this paper should cover the entire field of pipe flanges, as might be inferred from the scope this discussion has taken. Pipe flanges and their bolts must

* Author's closure, under the Rules.

necessarily be proportioned to the pressures and uses they are subjected to, and no universal and single standard can ever be constructed. My purpose was simply to take the existing practice of so-called high-pressure valve manufacturers and analyze the relation of pressures to bolt power. I have found that a gasket 1 inch wide, whether placed in a tongue and groove joint, or in a male and female joint, or in an absolutely plain-faced joint, will receive about 1,000 pounds per square inch of pressure from the bolts now provided for all sizes of pipes; that the width of gasket should not vary with the size of pipe, as is commonly supposed it should, for that would result only in pinching the gaskets more or less, while a uniform width of 1 inch would produce nearly uniform pressure for all sizes. In order to effect this result the better I suggested a few changes in dimensions of both, as is pointed out in the paper. I did not even intend to give my approval to the existing dimensions as best suitable to 250-pound pressures. On the contrary, I am not satisfied with them, but I intended only to call attention to the fact that an inch-wide gasket, or a gasket of any uniform width for all sizes of pipes, would receive uniform pressure per square inch of area from bolts proportioned as given in my table.

DCCCXVIII.*

STANDARDS FOR DIRECT-CONNECTED GENERATING SETS.

BY J. B. STANWOOD, CINCINNATI, O.

(Member of the Society.)

An important factor in America's industrial progress has been the extended application of standards. There are standard dimensions for materials; standard sizes, parts, and capacities of machines; standard gauges for close measurements; standard systems of inspection and testing, etc.

These different forms or methods of standardization have usually originated or developed in commercial and manufacturing activities, independent of any governmental instigation or control. They are universally recognized and employed wherever industry and commerce exist throughout our land.

This principle, as carried out in our individual shops and factories, has developed the well-known "interchangeable system" of construction, which, by forcing attention and care upon little things, has further specialized operations, and has stimulated the development of special machines to perform these operations, to the end that the cost of production has been greatly reduced.

Such standardization inside and outside of the shops also aids rapidity of production, inasmuch as standard material can be kept in stock to form standard parts or to complete standard machines, apparatus, or goods, all before these are even ordered.

These results—cheap production and prompt delivery—in connection with a high grade of product (to which these methods contribute) are most powerful factors for securing and maintaining foreign trade. Such influences are particularly active in America to-day.

In the steam-engine industry these methods have been largely in use; but within the past few years they have been rendered

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partially inapplicable by a radical change in the environment, due to the introduction of electricity and the employment of high steam pressure.

Whenever new types are undergoing development, before that weeding-out process known as "survival of the fittest" determines what the new type is to be, then standards have to wait until the process is complete. As regards the steam engine, it seems as if in one direction this determination of a type was wellnigh completed, and that the time for some standardization had arrived; and, the sooner that this is done, a source of great expense will be avoided, and opportunity for further development and progress extended.

By the introduction of an electrical generator attached to and directly driven by the main shaft of a steam engine a new machine has been evolved. But the introduction of this system has brought a world of complication and expense to both the engine and generator manufacturers, while, to some, serious financial loss has been the result.

The principal cause of these difficulties has been the lack of standard sizes, speeds, and important dimensions of electrical generators, so that the adaptation of the steam engine in each case has been made a special problem, frequently requiring special design and construction.

To complicate the matter, the determination of the sizes and speeds of such generators is usually in the hands of a third party, who is not in touch with either the engine or generator builder. He may be a consulting electrical or mechanical engineer, an architect, a salesman, an agent, or some person presumably skilled in deciding upon the necessary machinery to satisfy certain conditions.

If fortunately there had existed a list of capacities and speeds for electrical generators that was recommended by proper authorities, this list would gradually be used by such persons in securing and specifying the machine to be employed. As it now is, any odd size and speed may be selected by them; as, for instance, to my knowledge, a 45-kilowatt generator to be operated at 210 revolutions was once specified, almost at random.

If for such a generating "set" bids are taken, then the engine and dynamo contractors who are unlucky enough to secure the order must design a special outfit. Perhaps this is the first time that A's engine has ever been sold to drive one of B's

generators, and a new size of engine may have to be built, or *vice versa;* at any rate a correspondence ensues, with an interchange of drawings for a new pattern of sub-base which has to be made. It may happen that the engine shaft has to be enlarged to fit the armature, or the latter has to be changed to fit the shaft. Later the shaft has to be shipped to the dynamo shop to be pressed into the armature, or the armature sent 500 or 1,000 miles to be fitted to the shaft. In all this it is fortunate if mistakes or misfits do not occur.

When the combined plant is ready for erection the engine erector has to coöperate with the electrical erector; usually one waits for the other a few days, under expense, doing nothing. Then begins the adjustment of the engine's speed to suit the generator's speed, which may be slightly fast or slow, as the case may be; the governor, too, may be out in a similar way.

All told, these operations form a complicated and cumbersome system, which would be greatly ameliorated if there existed recognized and approved standards for a very few and simple conditions.

What are the features that need standardization? First, for the use of the outside engineer there is a need for a standard series of capacities and speeds; and, second, those parts of the engine and generator which have to be connected or fastened together should be standardized for each different size to facilitate the assembling of the combined machine or "set."

1. (*a*) Relative to a standard series of capacities and speeds, the different capacites or sizes should be selected to satisfy the usual demands of the market. The number of different sizes should not be too large, in order that the cost of production may not be enhanced by too great a variety of both generators and engines.

In this connection, there might be a standard method of determining the capacity of a generator, which would designate, with a given load or percentage of overload, a given time limit within which a permissible increase of temperature of the field-coils, armature, and commutator can occur.

(*b*) Relative to speeds, there should be at least three classes of speeds for each size or capacity of generator, corresponding, in the main, to the three classes of engines now on the market. These are known respectively as slow, medium, and high speed engines. By arranging the generator speed in this manner most of the conditions arising in practice could be met. There would be

slow speed for factory, manufacturing, or street-railway service, which would be expensive, but durable and easily tended. A medium speed could be used for lighting-plants or conditions where lower first cost and a more limited space exist, but which will require closer attention and will probably be less durable. Lastly, the high-speed "set" would be required for limited space, low first cost, but it will require close attention and will probably be subject to heavier repairs.

Perhaps a series of generators could be arranged so that, by a modification in windings only, a given size for a standard slow speed could be used for larger standard sizes, at standard medium and high speeds.

2. As to the standardization of the dimensions of those parts of the engine and dynamo which are connected together there are: (*a*) The shaft diameter and armature bore, which should both be expressed in thousandths of an inch; (*b*) the length of shaft outside of the engine, and the location thereon of the outboard bearing, should be given in feet and inches; (*c*) the distance from centre of armature to sub-base, in inches; and (*d*) the length and width of generator base, and the size and location thereon of the bolts which attach it to the sub-base, are probably all that are required.

Such a system would materially reduce the complication of construction and erection. A fewer number of engines with their sub-bases and generators would be required. These, in the marketable sizes, could be carried in stock ready for any combination, thereby permitting manufacturers to avail themselves of multiple production, so great a factor in reducing costs, improving quality, and facilitating delivery.

An American system of standards well introduced abroad and at home might give to us a large share of the electrical and engine business of the world.

In this connection, it seems to me that our Society and the American Institute of Electrical Engineers are the proper authorities to undertake such an investigation as will, by their recommendation, bring about such a result.

To initiate a movement for this purpose, I move that the Council be requested to communicate with the American Institute of Electrical Engineers to ascertain if that institute will agree to appoint a committee to coöperate with a similar committee to be appointed by the American Society of Mechanical Engineers, to consider and report upon a standard series of capacities, speeds,

and necessary dimensions for electrical generators for direct connection to steam engines.

And, furthermore, if a favorable response be received, then the President be requested, with the concurrence of the Council, to appoint a committee of five to coöperate with the committee to be appointed by the American Institute of Electrical Engineers for the purpose herewith set forth.

DISCUSSION.

Mr. E. P. Roberts.—It is unquestionable that it is desirable to standardize wherever possible. Standards have been effected in many matters which were thought impossible, or, at least, impracticable, and whether or not it is practicable to standardize generator sets, and at least to the extent suggested by Mr. Stanwood, is, it seems to me, somewhat questionable. That it is desirable is, I believe, unquestionably a fact.

Only last week I was in the shops of a prominent engine builder, discussing a bed-plate which he had designed for a set being installed under our specifications. Owing to certain features in the generator it was necessary to make a special design of bed-plate, and one which, in the engine builder's estimation, as well as our own, was not desirable. At the same time in his shops were bed-plates for other makes of generators, all of which differed in general design. With some the generator frame was movable parallel to the shaft; in others at right angles, and in others it was bolted in place.

Therefore, if standardization can be effected, it certainly will be better for the engine builder, the consulting engineer, and the client; whether it would be better for the generator builder is somewhat of a question, as it will at least lessen the number of "talking points," whether or not it lessens the value of the machine.

Also it would be desirable if the various builders made each the same size unit, and at the same speed or speeds. At present the consulting engineer decides that a dynamo approximately a certain size is desirable. He finds that several concerns make such a size and at a speed which is, in his estimation, suitable; but that there may be manufacturers of first-class material, and whose bid he desires, who do not make such size or operate at such speed, consequently they are barred out. If they are allowed to bid on

what they do make, then the size of the engine suitable for the other is not correct, and there is more or less difficulty in making matters tie up; even if there is not friction due either to barring out, or to accepting something different from that which is specified.

Relative to the size of shaft, it is not seldom that the engine builder desires one of larger diameter than can be allowed by the generator builder, although this feature is not now as often met as it used to be. We, however, have had this trouble in our own experience quite recently. We have also had a generator builder send a ring gauge which he stated to be the size of the bore of their armature spiders, with instructions to make the shaft $\frac{15}{1000}$ larger. The probability, however, of obtaining such dimension with a gauge which could not be slipped over the shaft, and which necessitated inside calipering of the gauge and making a new gauge $\frac{15}{1500}$ larger, is certainly not very good.

Whether or not the armature should be placed on the shaft at the dynamo builder's, the engine builder's, or when erected, also often gives rise to serious friction, but specifications should always cover such points. I hardly see how standardizing will affect same, as long as the engines and generators are built by different contractors. The same is true relative to one contractor's expert being delayed, at time of test, by that of another. However, delays of this kind should be provided for in the specifications.

Mr. Albert K. Mansfield.—In rising to second Mr. Stanwood's motion with which he concludes his paper, I would like to say that it has become a very important subject, especially from the standpoint of the builder of steam engines. At the present time it seems as if a new generator or new form of old generator came to light about every week, more or less, and that the manufacturer of the generator does not consider as carefully as he might in what way it can best be adapted to the different engines in the market. It becomes a great burden to the engine builder to adapt his machine to all these varieties of generators; in fact, so much so that when we can use the same patterns on second installations that we have used once, we find that there is after all some little pleasure in this business. It is rare that that can be done. As to the matter of speed, I do not think that the electrician considers carefully enough, generally, that the engine which drives the electric motor is a reciprocating machine. The

motor is rotary. The reciprocating machine has limitations; the motor perhaps none as to the matter of speed, or at all events not in the same direction. I hope that this will result in a consideration of that branch of the subject which will have a tendency towards slower speed; at least I mean slower speeds than the higher speeds which are now used in direct-connected plants. Another feature that I think should be well considered is in the application of the word "self-contained." It seems to be a notion that it is necessary in a great many cases to unite the parts of an installation of this sort with castings which appear to be massive. The fact is that with or without such castings the union must depend upon a substantial foundation. I believe it might well be considered by a committee of the sort proposed whether it might not be recommended that a substantial foundation is union enough. If so, it will result in economy to the buyer and simplification to the manufacturer. The subject is an important one. It is growing very complex, and for that reason I take pleasure in seconding Mr. Stanwood's motion.

Prof. W. S. Aldrich.—This paper refers to radical changes in steam-engine practice brought on by environment. I should like to ask what guarantee we have that this change shall not keep going on and on. All feel that we know what ought to be done, but have very little idea as to how to bring it about. These changes in environment are going on continually; and while we can standardize details of engines and dynamos, it is certainly a difficult matter to attempt to standardize an aggregate unit, much more so to standardize a unit so complicated as the dual combination of an engine-dynamo unit. In pumping engines the steam and water mechanisms constitute one unit; and a similar combination has been variously attempted also by manufacturers of combined engines and dynamos as an aggregate unit. It may be that the time is coming when engine builders will be dynamo manufacturers, and when dynamo builders will be engine manufacturers, but certainly that time is not here yet.

Mr. Stanwood speaks of a type of steam engine which has become well-nigh completed in one direction. I should like to inquire what particular type he refers to.

With particular reference to the high-speed, direct-connected unit for the type of plant brought before this meeting in other papers—office buildings, department stores, and so on—the cause of these difficulties has been attributed practically to electrical

engineers. The electrical industries came to the front and found certain established sizes of steam-engine units which they did not seek apparently to conform to. What early established the sizes for electrical generators but the requirements for lighting service? Power service early in its career squarely set the sizes for its generators. But the number of lamps furnished the basis on which generators were sold for lighting, and this later settled into certain well-known kilowatt capacities for dynamos. In other words, the number of lights required of the generator has set the pace for the electrical industry, to which now it appears the engine builder must conform.

It is further brought out in this paper that there already existed, whether fortunately or otherwise, a list of sizes and capacities of steam engines to which that of the dynamo builder was not brought to conform. It appears that the engine builder must now change all his designs, patterns, and standard sizes, and readapt them to the new conditions that he is brought face to face with in the electrical industry. All will recognize this difficulty. In the case of steam pumps, however, the builder, as we have noted, solved the whole problem himself. In the case of the engine-dynamo unit it is scarcely to be hoped that any one builder will solve the problem, but if he did that would bring this combination also rapidly to the front as a successful standardized aggregate unit.

Attention should be called to the points about overloading and the limits of expansion in these combined units. It is important to know at what rated capacity you are going to install the engine for a given dynamo unit. The dynamo, it is understood, will take a certain overload, 33⅓ to 50 per cent. for five or ten hours with a given rise of temperature. The engine is expected to meet that overload, of course, by increasing its expansion, as the author seems to have in mind the American automatic cut-off type and not that of the British throttle-regulated direct-connected unit.

A curious feature in dynamo designing is to be seen in the armature shafts, which have been up to the present time very small for the power transmitted through them. It is but necessary to call attention to the large size of the engine shaft of direct-connected units and the small size of the armature shaft to see the difficulty of standardizing a coupling. The sizes of shafts should be standard for both engines and dynamos.

Another point in Mr. Stanwood's paper is that of arranging a

series of windings on dynamos to be suited to the different capacities, "so that, by a modification in windings only, a given size for a standard slow speed could be used for larger standard sizes at standard medium and high speeds." I am afraid that our electrical designers will very seriously object to that. The work of throwing on and off coils on the field magnet of the dynamo to adapt it to changes of speed is not an everyday, an every hour, or an every minute question. It concerns the most vital question in regard to the design of the magnetic circuit as well as the effective working of the machine. It is not possible in the keen competition of the electrical business to change the field coil windings of the dynamo to suit the size of the engine. There might be possibly a partial solution of the mechanical difficulties involved in this question by having standard sub-bases or floor plates with T slots, so placing the smaller engines and dynamos on one bed-plate, as in the larger machine shop erecting floors.

Mr. W. D. Forbes.—The woes suggested by this paper and by the discussion are everyday, disagreeable, and, I am sorry to say, expensive facts to me. Therefore I take a great deal of interest in the paper. It seems to me, however, that the troubles suggested by Professor Aldrich and by others here are easily overcome. The question of the higher speeds I do not think need trouble us. I believe we can to-day produce engines commercially—not mere freaks, but commercial engines—that run at very much higher speeds than anything now to be found. I feel quite sure that if anything is to be done in this matter it should be done quickly, it should be done at once; because I happen to know that there is a strong effort in two foreign countries to do something in this line in order to obtain a better hold on the trade there, and it behooves us as engineers not to be behindhand in the matter. But the question of the sub-base which several of the gentlemen have spoken about is a very small matter. I think I can show plenty of designs, and there are others who can show designs, in which, with very small expense, changes in the sub-bases can be effected and the combination of the dynamo and the engine readily made in good shape; there is little or no trouble about that. But a trouble I do constantly find is that the dynamo makers do not seem to know, or at least are not willing to tell, the absolute truth about the power it takes to drive their dynamos. It is all right for these gentlemen; an owner goes ahead and overloads a dynamo, and it burns out, and the trick is done and there is no

trouble about who is to blame. But the engine is overloaded. Instead of running, it bumps around in all sorts of shapes, won't govern, simply because it is a boy trying to do a man's work. Here the dynamo man is all right, but the engine side is all wrong, and it is an important thing to find out, at once, how much power it takes to do the work which they say the dynamo will do. I find in making some tests that there is no comparison at all apparently in various makes of generators. I buy a 4-kilowatt machine from one man and a 4-kilowatt machine from another, and the amount of power taken to drive the two machines is totally different. Professor Denton is making some tests in this direction with me now. There does not seem to be any data on which they all agree as to how much power is needed to turn an armature around. I hope this matter will be taken up, for the reason, as I say, that there are two foreign nations which are attempting to do something in this line in order to get the trade which they now do not possess.

A Member.—Something has been said upon the engine side of the question, and I would like to add a few remarks on the other side, that of the dynamo builder. I think the dynamo builder appreciates as keenly as the engine builder the desirability of standardizing these two pieces of machinery, and while it is a problem which is extremely difficult, I think very much can be done, especially so far as standardizing the sizes and to a certain extent the speeds, from the dynamo builder's standpoint. In general, the higher the speed the better the operation of the machinery. The speeds have a given output; cannot be uniform; but there must be a slight variation on account of the various frequencies that it is necessary to use in electrical work. We use frequencies of 25 cycles, and 30, 40, and 60 cycles, and as the poles must be an even number, and for mechanical reasons it is necessary to have the number of poles divisible by four, it will be seen that with these various frequencies the same speeds cannot always be used. But if the engines can be designed so that they would vary in speed perhaps 5 per cent. above the normal rating and 5 per cent. below, all the conditions might be met for the various frequencies. With direct-current machines it is less difficult; because almost any engine can be used, and the machine can be built for it. It seems to me that if the speeds and the types of the generators and engines could be standardized, the other matters would be comparatively simple. In regard to

the sizes of shaft, I do not suppose that we can always retain those the same, because it is necessary to have different flywheel effects, and it is possible that the length of the machine and the place occupied by the flywheels could not always be retained the same for different machines. But no doubt a great deal can be done in this direction. Perhaps it might be accomplished by making the standard shaft a little heavier and a little longer than would be necessary for some of the cases; because we can afford to put a little more material into our apparatus provided we do not have to make a special design each time. I think the size of shafts might be made in even inches, and the size could certainly be determined once for all and a size adopted.

In regard to the capacity of dynamos, there ought not to be any difficulty, because it is certainly well known what the required power is to drive a dynamo with a given output, because the efficiencies of dynamos have been very accurately determined, so that we know the power required for driving within one or two per cent. Of course the question of ability to overload a dynamo is indeterminable, the same as with a steam engine, because it will be overloaded the same as a steam engine by varying the cut-off on the steam engine.

I think it is one of the most important questions of standards that have come up lately, because a very large per cent. of steam engines at present being built are used for dynamo driving; and the fact that dynamo builders have to build from many different specifications from various engine builders makes it fully as important from the electrical standpoint as from the mechanical standpoint. I hope that immediate steps may be taken to standardize this work as far as possible.

The President.—Is there any further discussion? If not, the motion of Mr. Stanwood is in order.

Mr. Stanwood's motion was read by the secretary and passed unanimously.

The Council subsequently gave favorable consideration to the subject, and under their action a committee, consisting of Messrs. Stanwood, Ball, Forbes, McFarland, and Rohrer, was appointed by the President.

DCCCXIX.*

BOILER AND FURNACE EFFICIENCY.

BY R. S. HALE
(Associate Member),
AND W. B. RUSSELL, BOSTON, MASS.

THE studies of boiler efficiency hitherto published have been mainly based either on theories that could not be satisfactorily checked by experiment, or else on experimental determinations of the efficiency without observing the facts necessary as a basis for a satisfactory theory. The reason for this is that in most experiments only the efficiency of the boiler as a whole has been observed, while this depends on various factors, such as the efficiency of the furnace, of the heating surface, of the grate, etc., etc.; and comparison of boiler efficiencies as a whole, without observing the furnace and other efficiencies separately, is only one degree better than would be a comparison of boilers based on coal used per horse-power hour indicated in an engine without determining the efficiency of the engine.

The following definitions bring out clearly the differences between the various efficiencies of a boiler.

Definitions.

The efficiency of a boiler, as a whole, is the ratio between the useful heat in the steam and the total heat originally in the fuel. It may be considered as being the product of three other efficiencies, the first of which is the furnace efficiency, or efficiency of combustion. It is the ratio between the sensible heat generated by combustion and the total heat supplied, the loss being the loss due to incomplete combustion.

The second is the efficiency of transmission, or of heating surface. It is the ratio between the heat transmitted to the steam

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and the sensible heat generated in the furnace, the loss being the loss due to the heat in the waste gases.

The third is the efficiency of utilization, or ratio between the heat utilized and the heat transmitted through the heating surface, the loss being the loss due to radiation.

These definitions are due to Hudson. (See London *Engineer*, 1890, p. 523.)

So far as we know, no theory governing the loss by incomplete combustion has ever been stated in exact form, but in 1897 one of the writers, in studying the efficiency of boiler heating surface, showed that the law governing the efficiency of heating suface as stated by Rankine could be most simply expressed by the formula :

$$\text{Efficiency of heating surface} = 1 - tcf - \frac{\frac{ac^2f^2w}{k^2s}}{1 - \frac{tcf}{k}}$$

in which t = temperature of steam above that of entering air ;
c = specific heat of gas ;
f = pounds gas per pound fuel ;
k = heat units generated per pound fuel ;
a = a constant ;
w = total heat units transmitted to steam ;
s = water-heating surface.

(See *Transactions* A. S. M. E., 1897, pp. 331 and 340.)

This formula shows that if the efficiency of combustion, the quantity of gas per pound fuel, and the term a, are constant, and the radiation negligible, the results of boiler tests when plotted, with efficiencies as ordinates and rates of evaporation as abscissæ, should fall on a straight line; and in the study of the efficiency of boiler heating surface above referred to it was shown that the straight-line formula indicated as well as any other the actual results of experiment. (See *Transactions*, vol. xviii., p. 339.) It was also shown that individual tests varied from the formula; but since individual variations might be due either to variations in the efficiency of combustion or to variations in the air supply, and did not necessarily imply error in the formula, it was thought that further investigation might prove of value if based on tests in which the efficiency of combustion and the air supply had been measured.

Tests.

For this purpose most of the tests given in the former paper were unavailable, for the reason that neither the efficiency of combustion nor the air supply was measured except in Professor Kennedy's tests of the Thorneycroft boiler. The tests given in Table I. have therefore been collected. In this table the first column is the reference number of the test. The second column gives the type of boiler, all of which types are familiar, except the German combined boiler, or double boiler, of which no cut is given. The third column gives the type of furnace. The next seven columns give the heat balance as given in the original tests, except in a few tests where it has been recalculated. The heat balances are not all figured in the same way, and although the difference is slight it should be noted. In the American tests the total heat in the fuel to which the other quantities are referred is the total heat theoretically available for heating water, including all of the latent heat of the H_2O in the gases, since this latent heat theoretically could be utilized if the object were to heat water and not make steam. In the German tests, on the other hand, the total heat in the coal is taken as the total heat theoretically available for making steam. If the coal is burned under a steam boiler, the hydrogen in the coal burns to gaseous H_2O, and develops approximately 52,200 British thermal units per pound. If the H_2O thus formed should condense it would give up another 10,000 British thermal units; but this additional heat is not available for making steam, and is therefore in the German tests omitted from the heat of the coal, since the comparison is of steam boilers and not water heaters. Similarly, the moisture in the coal must be evaporated before any steam can be made, and the heat necessary for this is, in the German tests, also subtracted from the heat of the coal. In the English tests the total heat of the coal is taken as the total heat in the dry coal theoretically available for making steam, and therefore omits the latent heat of the H_2O from the hydrogen, but makes no allowance for the latent heat of the moisture.

A second difference in presentation depends on the unburned coal in the ashes. The German tests use the total heat in the coal as fired, *i.e.*, they figure on coal. The American and English tests use the total coal burned, subtracting the coal found in the ashes, *i.e.*, they figure on combustible.

Having thus explained a slight difference in method of presentation of the heat balances as a whole, we will continue with the separate items.

Column 4, the first column of the heat balance, gives the heat in the steam.

Column 5 gives the heat in the hot gases. In cases where this has been recalculated, the weight of dry gas and specific heat of .25 have been used. The weight of total gas, including the H_2O, exceeds the weight of dry gas by some $\frac{1}{3}$ pound in about 20 pounds, or $1\frac{2}{3}$ per cent., and the H_2O has a specific heat of double that of the dry gas, making an error of about $3\frac{1}{3}$ per cent. In addition, the heat capacity of the moisture in the air is not measured or allowed for in any of the tests. Against these two errors is set the fact that the true specific heat of the dry gas is more nearly .24 than .25, an error of 4 per cent., so that these errors practically balance for ordinary air supplies. For very large amounts of gas the heat capacity thus computed would be slightly greater than the truth, and for small amounts slightly less, but considering the gain in simplicity of this method of computation over the exact methods, and the fact that the true specific heats are not very exactly known, it is thought to be justified.

Column 6 gives the heat loss by the formation of CO in the tests in which it was tested for and found appreciable. In many of the tests the loss from CO has been reported as zero on inadequate tests, so that it has been thought best not to make any difference in statement between the tests in which it was not measured, and the tests in which it is reported zero.

Column 7 gives the loss in the ashpit. In the American tests this item does not appear, since the heat of the coal lost in the ashpit is not included in the heat to which the heat balance is referred. In the English tests the same rule holds, the percentage given in some of the English tests referring to the sensible heat in the coal and ashes withdrawn from the grate at the end of the test.

Column 8 gives the loss due to latent heat of the H_2O in the gases. In the German tests this does not appear, since this heat was not originally considered as part of the total heat of the coal. In the English tests it includes only the latent heat of the moisture in the coal. In the American tests it includes all the latent heat of the H_2O in the gases.

Column 9 gives the radiation. In tests 78 and 79 this was

computed from the temperature and known rate of flow of heat through the boiler covering; in test 11 this was measured by the fall of pressure method; in the other tests (12 to 19 inclusive and 21), by noting the quantity of coal necessary to keep up the steam pressure without making steam.

Column 10 gives the unaccounted for loss. It includes the loss by hydrocarbons in the gases, and also all the other losses when not separately measured.

Column 11 gives the efficiency of combustion, *i.e.*, the ratio between the heat generated as sensible heat and the total heat theoretically available, and is the sum of columns 4 and 5. Strictly speaking, the true efficiency of combustion is higher than these figures by the amount of the radiation, but since the radiation is so seldom measured the method used will be found more convenient. It should also be remembered that this figure varies according to whether the German, English, or American method of measuring the heat in the coal is used, but the variation is unimportant for the present purposes.

Column 12 gives the efficiency of transmission. It is the ratio of column 4 to column 11, except in a few cases where the heat balance is given as originally reported, but column 12 recalculated. The neglect of the radiation introduces a slight negligible error into this figure also.

Column 13 gives the rate of transmission of heat through the heating surface in British thermal units per square foot per hour.

Column 14 gives the quantity of gas per 1,000 British thermal units of sensible heat, proportional to quantity $\frac{f}{k}$ in the formula. (When two figures are given the second refers to the quantity of gas measured at the bridge wall.)

Column 15 gives the rate of combustion in pounds of coal per square foot of grate per hour.

Column 16 gives the efficiency of transmission as calculated from the formula, using for a a value of 150. (When two figures are given the second is based on the quantity of gas at the bridge wall.)

Column 17 gives the difference between the calculated and observed efficiencies of transmission, being + when the observed value is greater than the calculated, and − when it is less.

Column 18 gives the kind of coal used.

Column 19 gives the number in the original reference when

necessary, and column 20 gives the original reference where the complete data may be found.

In selecting tests, practically all tests found giving the required data have been selected, except that no test in which over 100 per cent. of the total heat of the coal has been accounted for has been admitted. An over-accounting for the heat may be due to priming, and may be due to accidental error. In the latter case it would not be correct to reject the results in making up an average unless the similar errors in the opposite direction were also rejected, but since priming errors are all in the same direction, and have not been allowed for in many of the tests, and since they are in a direction opposite to any error that would arise from the rejection of these tests, it is thought that their rejection is justified.

No test has been used unless amount of gas has been determined by analysis, as anemometer and econometer tests were not considered sufficiently reliable.

Method of Analyzing Table I.

The tests presented in Table I. give a large amount of data, but it is obvious that the variations of single tests are so great that only a method of averages can be used to draw out useful information. This method is correct only if carefully used, but is the only method available. While some of the results given below may be considered as adequately proved, notably the fact that radiation is, in well-covered boilers, a much smaller loss than has often been reported, yet many of the results must be considered only as indications to be confirmed or reversed by future experiments.

The first question to be investigated will be the unaccounted for loss (column 10). This is generally ascribed to radiation, in which case the actual loss in heat units should remain constant whatever the rate of working, and in consequence the loss expressed as a percentage should decrease when the boiler is forced. Table II. divides the tests into five classes, according to the rate at which the boilers are worked, and the average per cent. unaccounted for (including radiation when separately measured) shows no tendency to decrease as the rate of working increases. Before accepting this result one or two possible sources of error will be investigated. It is theoretically possible that an increased loss

by incomplete combustion of hydrocarbons at high rates of combustion might have balanced the decreased percentage loss by radiation; but Table III., in which tests are divided into classes according to the rate of combustion, shows that if anything the higher the rate of combustion the less the unaccounted for loss up to the limit of 35 pounds per square foot of grate per hour. The high loss above this limit is probably due to particles of coal carried up the chimney by the strong draft necessary to burn coal at these high rates. (W. F. M. Goss's tests showed how this loss increases at high rates of combustion. See *Engineering News*, September 24, 1896.)

Table IV. is similar to Table II., but is limited to one class of boilers. It shows that the uniformity of the unaccounted for loss in Table II. was not due to averaging different types of boilers in one table.

These results show either that radiation is the same percentage, and consequently three or four times as much in heat units when the boiler is forced as when it is easily worked—a state of things inconsistent with any ordinary form of radiation—or else that radiation forms but a very small portion of the unaccounted for loss. That the latter is true may be shown in another way, since radiation may be determined independently of the unaccounted for loss by experimental methods of more or less accuracy. If the radiation is computed from the experiments on transmission of heat through boiler coverings, it is found to be only about 1 per cent. of the heat used at ordinary rates of working. (Tests 78 and 79 gave about 1 per cent. when calculated from the figures given in Norton's tests of similar boiler coverings. *Transactions* A. S. M. E., vol. xix., p. 729. See also A. S. M. E., vol. xviii., p. 917.) Another way of determining the radiation is to get up steam on a boiler, then draw the fires, closing all dampers, and note the time necessary for a certain amount of heat to radiate, as shown by the fall of steam pressure. The result determined by this method would naturally be higher than the truth on account of the difficulty of absolutely closing the dampers, and yet test 11 gave only 4.7 per cent. out of a total of 15.2 per cent. not otherwise accounted for, and tests 78 and 79 gave about $1\frac{1}{3}$ per cent. out of 15.1 and 9.9 per cent. not otherwise accounted. Still another method would be to measure the amount of coal necessary to keep up pressure without making steam. This would also be very much greater than the radiation alone, on account of

the loss in the flue and by air leakage, and yet in Donkin's tests (12 to 19 inclusive and 21) this method by no means accounted for all the heat.

In *Engineering News*, February 23, 1899, p. 118, are given some experiments on a locomotive boiler which indicated a radiation loss that would correspond to only about 2½ per cent. on the basis of Tables II., III., and IV., although the boiler was only partially covered.

Reviewing this information, we find that the most accurate method, which is computation from tests on boilers and pipe-coverings, indicates a loss of radiation from well-covered boilers of about 1 per cent. at ordinary rates of working (3,000 British thermal units transmitted per square foot, or 12 square feet per horse-power), and that other experimental methods, as well as Tables II., III., and IV., fail to show that the loss is larger than this. While the proof may not be conclusive, the inference is clear that the radiation is only about 1 per cent. at ordinary rates of working, about 2 per cent. when boiler is half worked, or ½ per cent. when forced to double the ordinary rate. This leaves about 8½ per cent. still unaccounted for of the 9.6 average unaccounted for heat. Some carbon is undoubtedly carried up the chimney with the draft. This may be put down as ½ per cent. In some of the tests CO, and in some (German tests on coal dust) the loss in the ashes, was not determined. Allowing 1 per cent. for these, we still find 7 per cent. unaccounted for, and hydrocarbon gases in the flue are the loss left to which we can ascribe this 7 per cent.

It should be noted that a large hydrocarbon loss introduces an error into the methods of figuring the gas per pound of coal. This quantity is usually figured from the gas per pound of carbon and the percentage of carbon in the fuel, but since some 7 per cent. of the heat is lost in the form of hydrocarbons, it is clear that some proportion, perhaps about 4 per cent., of the carbon is not measured in the gas analysis, so that this method of computation is in error by about this amount. This error is, however, too uncertain to be taken account of in the present calculations.

Unaccounted for Loss, continued.

Next, the subject of what causes this hydrocarbon loss will be investigated. Table V. divides the tests according to the air supply per 1,000 British thermal units generated, all the tests

being included, and the air supply taken as that at the flue in all the tests, while Table VI. makes the same classification, but includes only the tests in which the air supply was measured at the bridge, or in which there was no probability of air leakage having occurred, *i.e.*, in tests on combined, locomotive, return tubular, marine, and Thorneycroft boilers.

Strictly speaking, the classification should have been according to the air supply per 1,000 British thermal units that could have been theoretically generated, or, better still, according to the ratio of the air actually used to the amount theoretically necessary for perfect combustion, but inspection shows that the correction would make but slight difference, and the table indicates clearly that the hydrocarbon loss is, if anything, smaller when the air supply is small. This is such a surprising result that the dependence of the CO loss on the air supply will be at once investigated.

Table VII. shows that the CO loss does depend on the air supply, especially when we note that, strictly speaking, tests 12 and 78 should come under the first column, since the high figure of gas per 1,000 thermal units in those tests is not due to the large air supply but to the excessive incomplete combustion, reducing the thermal units generated. If these tests were placed in the first column of Table VII. it would make the average losses run approximately in the order $2\frac{1}{2}$, $\frac{4}{10}$, $\frac{2}{10}$, $\frac{1}{10}$ per cent., as the air supply increases, showing that the danger of loss from CO appears when the gas per 1,000 British thermal units falls below 1.3 pounds, or approximately below one and a half times the theoretical amount needed for perfect combustion. Many of the tests, however, show that it is possible to reduce the air supply below this amount without forming any serious amount of CO or hydrocarbons (tests 1, 2, 3, 6, 55, 82, and others).

Table VIII. is self-explanatory. It shows that neither stokers nor coal-dust firing is effective in decreasing the unaccounted for or hydrocarbon loss, and while the down-draft grates show an excellent result, the number of tests is so few as to prevent our accepting the result as final with any confidence.

Table IX. indicates that the unaccounted for loss increases as the quality of the coal decreases, the exception being the high figure for the single test of anthracite coal and for the Welsh coal. It is probable that the anthracite and Welsh would be more accurately represented by 7 per cent., and that the difference is

about 4 per cent. between the best and worst coals. If the average efficiency of the boilers be taken at 70 per cent., this would mean about 6 per cent. greater evaporation per heat unit from good than from poor coals. [N. W. Lord's results indicate (*Engineering News*, February 16, 1899) about 4 per cent. difference between Pocahontas and Ohio coals. Circular 5 of the Mutual Boiler Insurance Company showed that Nova Scotia coal was about 15 per cent. less value in a boiler test than Cumberland, though only about 8 per cent. less in theoretical heat units, indicating about 7 per cent. more heat unaccounted for.]

Western coals have been reputed to give lower efficiencies than these figures would show, and there may be some characteristic in Western coals that would have this effect. It is to be regretted that no boiler tests with Western coals were found in which both the coal analysis and gas analysis were given.

Table X. is self-explanatory. With the exception of the marine boiler tests, the cylinder boiler tests, and the test on a Lancashire boiler with gas generator, they show a superiority (by about 4 per cent.) for the boilers with fairly large furnace and brick walls over the boilers with small furnaces and no brick sides to the furnaces.

Cause of Hydrocarbon Loss.

The following hypothesis seems to explain the unaccounted for loss, namely:

The loss by hydrocarbons is inversely proportional to the temperature of the fire.

On reviewing the tables in the light of this hypothesis we find that in Table III. the temperature of the fire increases as the rate of combustion increases, for the reason that as the rate of combustion increases less time is offered for the coal bed to lose heat by radiation, so that the coal bed itself and the gases are hotter. As a result the unaccounted for loss (hydrocarbons) should and does decrease as the rate of combustion increases up to the point where the solid particles of coal are carried over by the draft. In Table V. the temperature of the fire theoretically increases as the gas per 1,000 British thermal units decreases, since the heat developed is taken up by a smaller amount of gas, and therefore raises it to a higher temperature, so that the hydrocarbon loss should decrease. The temperature, however, does not increase proportionately with the decrease of air supply on account of the

phenomenon of dissociation, so that the increase of temperature and decreased hydrocarbon loss at low air supply are but slight.

In Table VIII. neither stokers nor coal-dust firing would differ very much from hand firing in average temperature of the fire, while the down-draft grate with its double combustion should show a high temperature and low hydrocarbon loss.

In Table IX. the poor grades of coal with large amounts of oxygen, which reduces the temperature, and with a large proportion of their heat units in the form of hydrocarbons, naturally show a larger loss than the good coals.

In Table X. the boilers with brick in the furnace and the boilers with large furnaces naturally have a higher furnace temperature and lower loss than the boilers like the Lancashire or locomotive, in which the temperature of the fire is quickly reduced by radiation to the cold boiler shells close to the fire.

Since the loss is greater with poor coals it is natural to expect that good furnace conditions would be more important with poor coals than with good coals.

Efficiency of Heating Surface.

In regard to the efficiency of heating surface, the accuracy of the formula will be first investigated.

Table XI. shows the average errors of the formula at different rates of evaporation from the shell boilers (Lancashire, cylinder, etc.), for fire-tube boilers, for water-tube boilers of the Heine and B. & W. class, and for the Thorneycroft and water-tube boilers, while Table XII. shows the same results for all classes of boilers. Considering the probability of errors, the results show a gratifying agreement with the formula at all rates of evaporation, with the exception of the tests of water-tube boilers of the Heine and B. & W. type at low rates of evaporation. This exception is probably due to short circuiting of the gases at low rates of working and consequent ineffectiveness of a large part of the heating surface. The fact that the Thorneycroft, fire-tube, water-tube with large tubes, and Lancashire boilers all give results agreeing about equally well with the formula (with the single exception noted), indicates that the circulation, both of water and gases, is equally good in all those types.

Table XIII. compares the results of the formula when air

supply is varied. The results indicate that the formula gives too low results by about 1½ per cent. when the air supply is small, and too large results by about 2 per cent. when the air supply is large—the very large figure for air supply over 1.9 with double tests being most probably due to accidental error or air leakages after the gases have left the boiler. The figures with gases measured in flue only on boilers subject to air leakage naturally show results higher than the formula when the air supply, as measured, is large, since in many of these cases the large air supply is due only to leakage.

Part of this divergence from the formula is accounted for by the approximate method of calculation used (see p. 772). When the air supply is large, the error in neglecting the H_2O is less than the error from using .25 instead of .24 as the specific heat of the gas. This, however, would account for less than 1 per cent., leaving the formula about 1 per cent. too low at low air supply, and about 1 per cent. too high at large air supply. This divergence from the formula is clearly not due to any variation in the speed of the gases over the heating surface, since if it were due to the speed it would appear in Tables XI. and XII., where the variation in speed over the heating surface is over twice as great as the variation in speed in Table XIII. The variation from the formula is most probably due to air leakage into the flue after the gases have left the boiler, but before they are measured. Such air leakage would cause the formula to indicate a lower result than the truth, *i.e.*, would make the variation +, and would be more often present when the air supply, as measured, was high. It is probable, therefore, that the formula represents the average efficiency of heating surface within 1 per cent., except in certain boilers, where the gases short circuit at low rates of evaporation, and that variations from the results indicated by the formula are due either to experimental error or to variation in the condition of the heating surface, *i.e.*, to whether it is clean or covered with scale or not. The comparison of the unaccounted for losses with the variations from the formula will give a slight indication of whether experimental errors or variations in condition of heating surface are most frequent, and is given in Table XVI. An error in the steam measurement that makes a + variation from the formula diminishes the unaccounted for. An error in the gas that makes a + variation in the formula increases the unaccounted for. Now, if the gas measurements are all correct

and the steam measurements in error, we should find that tests with + difference from the formula usually have a small per cent. unaccounted for. If the steam measurements are correct, and the gas measurements in error, the reverse is true. If they are both correct, the tests with variation in the formula would be equally divided between the tests with small and the tests with large unaccounted for losses.

Now, Table XIV. shows no great tendency for the tests with either + or − variations to fall into either the class of low or the class of high per cents. unaccounted for, and since it is very improbable that the + error in the gas and the − error in the steam, or vice versa, should both be large, and should also balance each other, the inference is that the tests are reasonably correct, so far as the steam and gas are concerned, and that variations from the formula are due to variations in the condition of the heating surface (which would be caused by short circuiting of gases, or by scale or soot on the heating surface) rather than to experimental error. Of course, experimental errors are undoubtedly present, and are often large, but the table (XIV.) indicates that on an average they are not so important as the variations in condition of heating surface. It should also be noted that this comparison shows nothing whatever about the accuracy of the measurements of the fuel or the heat therein.

Air Supply.

The amount of air used in different types of furnace is shown in Table XV. The fact that mechanical stokers need fully as much air to develop the heat as hand firing is a surprise. The other figures are much what we should expect.

The variation in air supply with different rates of combustion is shown in Table XVI. The air supply decreases markedly as the rate of combustion increases.

It is, however, probable that it is the ratio between the square root of the draft pressure and the rate of combustion that affects the air supply, and not the rate of combustion alone.

The variation of air supply with various grades of coals is shown in Table XVII. The table gives no indication of any law.

The loss in efficiency due to air leakage is shown in Table XII. Boilers in which the gases were tested in both bridge and flue showed about 4 per cent. worse efficiency than that indicated by

the formula using the air supply at the bridge. Since the formula appears to be nearly correct, this is the measure of average loss by air leakage in these boilers, although it is much higher in some cases. Using Table XI., the loss appears to be about the same for the Lancashire types and for the water types subject to air leakage, after allowing for the loss in two of the water-tube tests by short circuiting of the gases.

The lessons to be learned from these data are as follows. Most of them only confirm present best practice:

In designing or selecting a boiler have the furnace roomy, and, if practicable, line it with brick in order to have a very high furnace temperature.

Pay especial attention to the furnace if poor coal is to be used. Get rid of air leaks after the gases have left the hot furnace. Proportion the boiler so that at least 15 pounds of coal per square foot of grate must be burned to develop its horse-power, and so to keep the rate of evaporation per square foot of heating surface low. This involves a moderately high ratio of heating to grate surface.

The horizontal return tubular boilers appear, all things considered, to be the type of boiler for best economy of steam production, although the internally fired boilers are probably as good if the furnace is roomy and the firing careful. The Thorneycroft boiler, which (from a *combustion* point of view) is of this type, shows the best result of all.

Certain of the water-tube boilers suffer from air leakage and from short circuiting of gases if underworked. Of course the advantages of water-tube boilers often make their choice advisable, even at the possible expense of a slight loss in efficiency.

The Lancashire type suffers from air leakage, and the furnace is poor.

Stokers save nothing over hand firing, either in better combustion or less air supply. This does not necessarily apply to all stokers, and it should be remembered that, even if it should finally prove to be true that stokers are no more efficient than hand firing, yet saving in labor and smoke may pay good returns on their cost.

Down-draft grates show up well, both in good combustion and small air supply, but the number of tests is too few to allow final conclusions to be drawn.

The results show that present knowledge enables the boiler

efficiency to be predicted reasonably well, provided the efficiency of combustion and the air supply are accurately known, and although these quantities themselves cannot as yet be accurately predicted, yet it is shown that the temperature of the furnace is at least one factor in the question of good combustion, and that rate of burning per square foot of grate either affects the air supply or is affected by some causes that also affect the air supply so that they vary together.

The most pressing need, therefore, in the study of boiler efficiency is the further determination of the factors that govern the efficiency of combustion and the air supply. The latter can be studied by the means of taking gas analyses at the bridge wall for different methods or frequency of firing, thickness of fire, etc. The factors governing efficiency of combustion can only be determined by carefully conducted and complete tests with heat balance. In this connection it should be noted that one or two *accurate* tests, in which all the quantities, such as thickness of fires, gas analysis, etc., are observed, are, for the purpose of increasing our knowledge of boiler efficiency, worth a thousand tests in which the efficiency or evaporation is measured alone.

TABLE I.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
			Heat Balance.							Effi-ci'ncies									
Reference Number.	Type of Boiler.	Type of Furnace.	Steam.	Hot Gas.	CO.	Ashes.	H_2O.	Radiation.	Unacco'nted for.	Of Combustion.	Of Transmissi'n and Utilization.	B. T. U. Transmitted p. Sq. Ft. of Surface per Hour.	Lbs. Gas p. 1,000 B. T. U. Heat Gen. in Furn.	Lbs. Coal per Sq. Ft. of Grate per Hour.	Efficiency of Transm'n Calcul'd fr. Formula.	Difference between Observed and Calculated Values.	Coal Used.	Number in Original Report.	Reference where Complete Data may be Found.
1	Thorneycroft water-tube	Ordinary	86.8	10.8	.5	....	....	....	1.9	97.6	88.	1,428	1.27	7.7	86.8	+1.2	Welsh		A. B. B. Kennedy, *Proc.* I. N. A.
2			81.4	15.	....	....	....	....	3.6	96.4	84.3	3,660	1.26	18.6	87.1	−2.8	"		
3			78.2	16.5	3.0	....	....	....	....	94.7	82.6	5,340	1.27	29.8	80.7	+1.9	"	....	Also pamphlet published by Thorneycroft & Co.
4			66.6	20.3	9.2	....	....	....	2.3 3.9	86.9	76.6	9,480	1.17	66.8	76.6	± .0	"		
5	Horiz. return tubular.	Regan (ord'y)	62.9	20.8	5.7	....	....	....	....	84.7	75.1	7,794	1.43	30.9	72.7	+2.4	Clearfield		Pamphlet published by Hawley Furnace Co., March, 1897.
6		Down-draft (Hawley)	79.6	17.3	.3	....	....	....	10.6 2.8	96.9	82.1	8,306	1.15	30.8	81.4	+ .7	"	....	
7	"	"	72.3	15.6	.0	...	3.1	....	9.	87.9	82.3	3,767	1.54	20.	78.3	+4.	Pocahontas	.....	Paper by R. S. Hale, *Trans.*, vol. xviii.
8	.....	Cok'g st'ker (Roney)	59.5	33.4	.8	....	....	....	6.3	92.9	64.0	1,401	2.65	9.2	68.6	−4.6	"	....	Theses at Mass. Inst. Tech.
9	H. ret. tub'r	"	70.	17.8	.0	2.2	....	....	10.	87.8	79.7	2,038	1.59	12.6	82.6	−2.9	Soft	.....	
10	B. & W.	"	71.2	14.3	.0	2.5	5.3	...	6.7	85.3	83.2	2,471	1.99	17.9	75.9	+7.3	"		
11	Lancashire.	Coal dust firing	65.9	18.2	....	.7	....	4.7	10.5	84.1	81.6	2,782	2.21	....	72.5	+9.1	Welsh	.....	Holliday, *Proc.* I. C. E., vol. xcii., p. 350.
12	Vert. tub'r.	Ordinary	63.8	9.4	12.7	.1	.1	4.5	9.4	73.2	86.3	2,251	1.52	17.9	85.2	+1.1	"	I.	
13	Lancashire.	"	70.4 3.4	13.6	....	.3	.4	7.	4.9	84.0	82.2	2,530	1.94	10.2	76.9	+5.3	"	II.	
14	"	"	67.6 4.6	16.2	1.2	.4	.4	6.0	3.6	83.8	80.5	4,616	1.38	18.5	82.7	−2.2	"	III.	
15	Loc'motive.	"	66.2	22.5	....	.2	.1	5.2	5.8	88.7	75.7	3,352	2.06	7.9	76.8	−1.1	"	IV.	Donkin's summary of twenty-one tests, *v. Engineering*, Sept. 20, 1895.
16	"	"	56.8	31.8	....	.4	....	9.4	1.6	88.6	60.9	1,662	2.98	6.2	60.0	+0.9	"	V.	
17	Cornish	"	78.3	14.0	1.7	.2	0.0	4.4	1.4	92.3	85.0	2,260	1.52	6.9	86.8	−1.8	"	VIII.	
18	Loc'motive.	"	69.8	18.0	1.2	.0	.1	5.7	5.2	87.8	79.5	3,920	1.47	12.4	82.1	−2.6	"	XI.	
19	Corn. mult'r	"	82.7	20.7	....	.1	....	8.2	8.3	83.4	74.2	2,434	2.30	13.7	70.8	+3.4	"	XVII.	
20	Cornish	"	61.8	20.4	....	.1	.1	....	17.6	82.2	73.3	7,176	2.25	15.7	47.9	+25.4	"	XIX.	
21	Corn. mult'r	"	65.8	18.0	1.6	.1	.1	8.	6.4	83.6	77.6	5,235	1.31	21.5	81.8	−1.2	"	XX.	

22	Cornish ...	Ordinary....	73.4	16.1	....	3.5	....	....	6.9	89.5	82.1	5,883	1.41	24.6	77.8	+4.3			I.		
22a	....	"	....	....	....	....	....	....	....	....	....	....	1.12	....	84.5	−2.4					
23	Inclined....	"	62.0	22.0	....	2.4	....	....	13.6	84.	73.8	3,500	2.56	25.	56.7	+17.1			II.		
23a	Water-tube.	....	....	....	....	....	....	....	....	....	....	....	1.51	....	81.5	−7.7					
24	"	Ordinary	69.9	22.6	....	2.	....	....	5.5	92.5	75.6	4,570	1.93	20.9	68.6	+7.0			III.		
24a	....	....	....	....	....	....	....	....	....	....	....	....	1.41	....	81.0	−5.4	Ruhr coal; about 78% of fixed carbon in combustible.				
25	Vertical wtr. tubes	Ordinary...	71.0	20.8	....	.6	....	....	7.6	91.8	77.3	3,140	1.70	18.5	79.2	−1.9			IV.		
25a		....	....	....	....	....	....	....	....	....	....	....	1.57	....	81.5	−4.2					
26	Inclined wat'r tube	Ordinary...	70.0	17.7	....	1.7	....	....	10.6	87.7	79.8	3,470	1.76	22.2	76.2	+3.6			V.	Frankfort a/M. Exhibition, 1891; vol. ii. of *Report*.	
26a		....	....	....	....	....	....	....	....	....	....	....	1.48	....	83.7	−3.9					
37	"	Ordinary.....	73.3	17.7	....	3.1	....	....	5.9	91.	80.6	3,360	1.49	12.6	81.9	−1.3			VI.		
27a	....	....	....	....	....	....	....	....	....	....	....	....	1.39	....	83.6	−3.0					
28	C'rnish.	Ord'y. with smoke prevent'g dev. (Air adm'n)	79.2	13.5	....	2.2	....	....	5.1	92.7	85.4	4,130	1.27	16.	84.3	+1.1			VII.		
28a			....	....	....	....	....	....	....	....	....	....	1.05	....	88.1	−2.7					
29	Marine	Ordinary.....	62.0	18.5	3.6	2.9	1.2	....	11.8	80.5	76.3	4,771	1.28	19.2	81.8	−5.5	Scotch	....	....	88. *Meteor*	*Proc.* British Institute Mechanical Engineers, 1890.
30	"	"	62.	28.0	1.3	1.7	.4	....	6.6	90.0	69.1	8,132	1.59	26.1	67.9	+1.2	Yorkshire	.....	.....	88. *Colchester*	
31	"	"	57.2	23.5	...	....	.9	....	8.4	90.5	74.4	3,747	2.02	19.0	70.9	+3.5	Tyne	.....	.....	88. *Fusi Yama*	
32	"	"	69.2	16.2	....	1.6	....	....	13.0	85.4	80.8	3,062	1.96	22.1	71.7	+9.1	"	.....	.....	88. *Iona*	
33	"	"	56.1	26.8	....	3.1	....	....	4.0	92.9	79.9	9,561	1.38	31.3	71.7	−0.8	Patent block ...	.....	.....	88. *Ville de Douvres*..	
34	B. & W....	"	79.2	15.5	....	.9	....	....	4.4	94.7	80.1	3,472	1.80	14.7	75.6	+4.5	Coke	.....	.....	No. 2 of 1889, *Zeit.* Dampfkessel- und Versich-verein, a. T. 1889.	
35	German double b'l'r	Coal-dust ..	76.8	13.4	....	....	....	....	9.8	90.2	85.1	3,990	1.51	....	80.1	+5.0	German Steinkohl		1		
35		Wegener...	79.3	13.4	....	....	....	....	7.3	92.7	85.5	4,615	1.47	....	80.	+5.5	"		2		
37	"	"	79.2	13.4	....	....	....	....	7.4	92.6	85.5	4,395	1.56	....	78.3	+7.2	Eng. Steinkohl.		3		
38	"	"	78.5	15.8	....	....	....	....	5.8	94.3	83.3	3,840	1.67	....	78.3	+5.0	Bohemian		4		
39	"	"	78.1	14.6	....	....	....	....	7.3	92.7	84.3	2,980	1.63	....	81.4	+2.9	brown coal.		5		
40	Locom'tive.	"	70.6	10.8	....	4.7	....	....	13.8	81.4	86.7	3,460	1.37	....	85.2	+1.5	Steinkohl		6		
41	"	"	73.6	9.6	....	....	....	....	16.8	83.2	86.9	2,810	1.39	....	86.1	+1.3	"		7		
42	Lancashire.	Coal-dust	79.6	11.1	....	2.7	....	....	6.6	90.7	88.	4,135	1.36	....	83.8	+4.2	Upper Silesia ..		1		
42a	....	....	....	....	....	....	....	....	....	....	....	....	1.19	....	86.8	+1.2	....		....		
43	Lancashire.	Schwartz-kopff	78.4	10.9	....	2.7	....	....	8.0	89.3	87.8	4,125	1.15	....	87.5	+0.3	Upper Silesia ..		2		
43a	....	....	....	....	....	....	....	....	....	....	....	....	0.98	....	90.2	−2.4	....		....	1895: *Report* of Dampfkessel Revisionsverein, Berlin.	
44	Lancashire.	....	71.4	15.1	....	2.7	....	....	8.1	89.2	82.9	5,340	1.24	....	84.0	−1.1	Upper Silesia ..		3		
44a	....	....	....	....	....	....	....	....	....	....	....	....	.99	....	88.6	−5.7	....		....		
45	Lancashire.	....	72.2	10.4	....	2.5	....	....	15.0	82.6	87.4	4,325	1.16	....	87.0	+0.4	Westphalian ...		4		
45a	....	....	....	....	....	....	....	....	....	....	....	....	1.02	....	89.8	−2.4	....		....		
46	Lancashire.	....	72.8	12.7	....	2.7	....	....	11.9	85.5	85.2	4,880	1.19	....	85.0	+0.2	Upper Silesia ..		1		
46a	....	....	....	....	....	....	....	....	....	....	....	....	1.12	....	86.4	−1.2	....		....		
47	Lancashire.	Coal dust ..	77.1	11.7	....	2.6	....	....	8.5	88.8	86.7	4,125	1.14	....	86.3	+0.4	Upper Silesia ..		2		
47a	....	....	....	....	....	....	....	....	....	....	....	....	1.12	....	89.4	−2.7	....		....		
48	Lancashire.	Friedeburg.	66.1	13.4	....	2.6	....	....	18.0	79.5	83.2	5,530	1.12	....	85.1	−1.9	Upper Silesia ..		3		
48a	....	....	....	....	....	....	....	....	....	....	....	....	1.12	....	87.4	−4.2	....		....		
49	Lancashire.	....	73.4	12.4	.	2.5	....	....	11.7	85.8	85.5	4,525	1.12	....	87.6	−2.1	Westphalian ..		4		
49a	....	....	....	....	....	....	....	....	....	....	....	....	1.02	....	90.5	−5.0	....		....		
50	Cornish ...	Ordinary.....	75.5	14.7	....	1.4	....	....	8.4	90.2	83.7	5,680	1.1	18.2	83.3	+0.4	German, 78% fixed carbon.		.....	Prussian Dampfkesselverein, May 1, 1897.	
51	" ...	"	70.8	16.7	....	1.8	....	....	10.7	87.3	80.9	8,020	.99	27.8	85.0	−4.1					

TABLE I.—*Continued.*

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
			Heat Balance.							Efficiencies									
Reference Number.	Type of Boiler.	Type of Furnace.	Steam.	Hot Gas.	CO.	Ashes.	H_2O.	Radiation.	Unaccounted for.	Of Combustion.	Of Transmission and Utilization.	B. T. U. Transmitted per Sq. Ft. of Surface per Hour.	Lbs. Gas p. 1,000 B. T. U. Heat Gen. in Furn.	Lbs. Coal per Sq. Ft. of Grate per Hour.	Efficiency of Transm'n Calcul'd fr. Formula.	Difference between Observed and Calculated Values.	Coal Used.	Number in Original Report.	Reference where Complete Data may be Found.
52	Cornish		78.5	10.4	....	....	....	....	11.1	88.9	88.6	4,100	1.25	....	86.3	+2.3	English.....	.....	
52a	.....		.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.99	....	90.2	−1.6			
53	Cornish	Camp coal-dust.....	78.5	10.4	....	....	....	....	11.2	88.9	88.5	4,325	1.00	....	88.2	+0.3	English.....	.....	Prussian Dampfkesselverein, Feb. 15, 1879.
53a	.....		.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.93	....	90.8	−2.3			
54	Cornish		74.9	11.4	.1	....	....	....	13.6	85.3	87.0	4,490	1.15	....	87.2	−0.2			
54a	.....		.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.95	....	90.4	−3.4			
55	Cahall	Ordinary.....	78.7	11.8	2.6	....	3.5	....	3.4	90.5	87.0	3,550	1.02	16.3	90.8	−3.8	Pocahontas	.....	Pamphlet by Thayer & Co., Aug. 9, 1897.
56	B. & W.	"	72.6	12.4	2.2	....	....	....	12.7	85.0	85.4	3,450	1.40	20.9	82.8	+2.6	Welsh	.....	Pamphlet by H. Robinson, London, 1895.
57	Lancashire	Ordinary / Kourtyke (Air Admis'n)	76.5	17.8	.0	.4	....	....	5.3	94.3	81.1	4,749	1.79	24.4	74.6	+6.5	Oberschlesische	I. 1	
57a	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.57	....	79.6	+0.5			
58	"	"	80.3	14.6	....	.4	....	....	4.7	94.9	84.6	4,958	1.53	24.4	80.3	+4.3	.....	I. 2	
58a	.....	"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.33	....	84.2	+0.4			
59	Lancashire	"	73.6	18.7	....	.6	....	....	7.1	92.3	79.8	3,706	2.17	28.1	71.4	+8.4	Bohemian br. coal.....	I. 3	
59a	....	"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.88	....	77.5	+2.3			
60	Lancashire	(Ordin'y) / Chubb ...	71.0	18.3	...	2.8	....	....	7.8	89.3	79.5	4,519	1.88	15.7	71.9	+7.6	Oberschlesische	II. 1	
60a	.....	"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.70	....	76.3	+3.2			
61	Lancashire	"	69.7	18.9	....	3.0	....	....	8.3	88.6	78.7	4,812	2.14	18.3	63.4	+15.3	Oberschlesische	II. 2	Rauchverbrennung Commission *Bericht*, Berlin, 1894.
61a	.....	"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.89	....	70.5	+8.2			
62	Lancashire	"	68.5	22.1	....	1.1	....	....	8.3	90.6	75.6	6,904	1.59	24.9	72.2	+3.4	.....	II. 3	
62a	.....	"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.42	....	77.0	−1.4			
63	Lancashire	"	62.4	23.4	....	2.2	....	....	12.0	85.8	72.8	5,816	1.82	34.5	67.7	+5.1	Bohemian br. coal.....	II. 4	
63a	.....	"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.53	....	75.9	−3.1			
64	Lancashire	(Ordin'y) / Strauss ..	71.2	18.6	.2	1.1	....	....	8.8	89.9	79.3	6,822	1.66	23.4	69.5	+9.8			
64a	.....	"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.48	....	75.0	+4.3	Oberschlesische	III. 2	

65	Lancashire.	"	7.06	20.4	1.5	1.2	....	....	6.2	91.0	77.6	4,610	2.10	15.7	64.6	+13.0	Oberschlesische	II. 2			
65a	....	"	....	....	....	....	....	....	....	....	....	....	1.85	....	71.3	+6.3					
66	Lancashire.	"	60.6	21.8	.6	1.1	....	....	15.9	82.4	73.5	7,499	1.77	39.4	63.8	+9.7	"	III. 4			
66a	....	"	....	....	....	....	....	....	....	....	....	....	1.46	....	74.0	−0.5					
67	Lancashire.	"	57.0	26 0	1.2	2.8	....	....	13.1	83.0	68.7	6,287	1.92	26.1	63.2	+5.5	English	III. 5			
67a	....	"	....	....	....	....	....	....	....	....	....	....	1 51	....	75.5	−6.8					
68	Cornish	(Ord.) Kuhn..	64.5	22.4	2.4	.4	....	....	10.3	86.9	74.5	2,952	2.16	23.6	68.3	+6.0	Mixed	IV. 1			
68a	....	"	....	....	....	....	....	....	....	....	....	....	1.62	....	79.7	−5.4					
69	Cornish	"	64.0	22.4	.4	1.1	....	....	12.2	86 4	74.1	4,821	1.64	43.7	73.9	+0.2	Bohemian b'wn coal.....	IV. 2			
69a	....	"	....	....	....	....	....	....	....	....	....	....	1.30	....	81.9	−7.8					
70	Cornish	"	66.0	24.4	.4	.2	....	....	9.0	90.4	73.0	4,624	1.56	29.5	76.3	−3.3	Oberschlesische	V. 1			
70a	....	"	....	....	....	....	....	....	....	....	....	....	1.25	....	83.2	−10.2					
71	Fire tube like an elephant	(Stoker) Tenbrink	70.8	18.7	.1	1.0	....	....	9.9	89.0	79.0	2,912	2.04	25.8	72.6	+6.4	"	V. 2	Rauchverbrennung Commission *Bericht*, Berlin, 1894.		
71a			....	....	....	....	....	....	....	....	....	....	1.80	....	77.2	+1.8					
72	return tubular	von Cyclop.	70.2	20.3	.1	1.0	....	....	8.4	90.5	77.6	2,308	1.84	27.4	78.5	−0.9	"	V. 2			
72a	....	....	....	....	....	....	....	....	....	....	....	....	1.52	....	83.5	−5.9					
73	Heine....	(Ordinary) Schomburg	58.1	31.8	.8	3.4	....	....	6.4	39.9	34.6	2,145	2.68	18.6	65.3	−0.7	"	VI. 1			
73a	....	"	....	....	....	....	....	....	....	....	....	....	1.95	....	78.6	−14.0	"				
74	Heine.....	"	62.4	23.3	1.3	3.4	....	....	9.5	85.7	72.8	2,239	2.01	18.4	76.5	−3.7	"	VI. 2			
74a	....	"	....	....	....	....	....	....	....	....	....	....	1.59	....	83.2	−10.4					
75	Lancashire.	Donneley	63.2	21.8	....	4.9	....	....	10.0	85.0	74.3	....	1.92	28.0	68.8	+5.5	"	VII. 1			
75a	....	(Stoker)	....	....	....	....	....	....	....	....	....	4,979	1.56	....	78.0	−3.7					
76	Lancashire.	" ...	72.4	14.8	....	5.0	....	....	7.7	87.2	83.0	5,396	1.47	26.5	78.8	+4.2	"	VII. 2			
76a	....	" ..	....	....	....	....	....	....	....	....	....	....	1.28	....	83.1	−0 1					
77	....	" ...	64.5	17.5	0.2	5.4	....	....	12.3	82.0	78.6	6,821	1 48	45.5	75.4	+3.2	Bohemian b'wn coal....	VII. 3			
77a	....	" ...	....	....	....	....	....	....	....	....	....	....	1.32	....	82.1	−3.5	"				
78	Vertical tubul'r.	Ordinary.....	51.5	9.0	18.9	2.7	2.9	1.1	14.0	60.4	85.1	2,520	1.34	16.0	85.3	−0.2	Cumberland ...	.....	Unpublished.		
79		"	61.8	17.0	3.0	5.4	2.9	1.2	8.7	78.8	78.4	2,880	1.62	15.3	79.5	−1.1	" ...	.....	"		
80	Cylinder ...	Coxe stoker..	69.3	10.2	.0	2.7	2.4	....	15.4	79.5	85.7	3,700	1.22	....	86.3	− .6	Anthracite	.....	N. E. Cotton Mfrs.' Ass'n, 1895, p. 65.		
81	B. & W..	Down-draft (Hawley)..	72.1	17.8	.0	....	2 7	....	7.9	89.4	80.7	4,390	1.86	31.0	69.3	+10.4	Pocahontas	.....	Unpublished.		
82	Horiz. return tub'r	Ordinary.....	76.1	13.4	.3	....	3.5	....	6.7	89.5	84.5	3,895	1.14	16.4	84.3	+0.2	" ...	.....	*Engineering Record*, June 4, 1898.		
83		"	75.1	14.3	.6	....	3.7	....	6.1	89.6	83.6	3,745	1.25	16.4	86.0	−2.4	Cumberland ...	.....			
84	Wood water tube	Murphy st'r..	74.3	17.3	1.5	....	2.2	....	4.7	91.6	77.7	2,891	1.78	14.7	79.3	−1.6	Pocahontas ...	.....			
85		Down draft. (Hawley)..	79.6	13.3	1.9	....	2.3	..	2.9	92.9	84.4	2,705	1.59	15.3	83.3	+1.1	"	.....	*Engineering Record*, April 30, 1898.		
86		Murphy st'r..	74.2	16.8	1.1	....	5.1	....	2.8	91.0	79.6	2,897	1.91	19.8	77.2	+2.4	"	.....			
87	L'motive.	Stoker (Schragrost)	76.1	14.8	....	6.4	....	....	2.7	90.9	84.7	3,394	1.40	28.3	84.7	±0.0	Mixed br. coal .	II.			
87a	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.....			
88	Galloway ..	Don'ley stkr..	65.9	16.5	....	1.4	....	....	16.2	82.4	81.5	2,913	1.54	20.4	82.0	−0.5	Brown nut coal.	V.	*Bericht* über Rauchfreie Dampfkesselanalegen in Sachsen, I. L. Lewicke, 1896.		
88a	"	"	....	....	....	....	....	....	....	....	....	....	1.28	....	86.3	−4.8					
89	C'mbined	Stoker (Schragrost)	67.9	15.7	....	0.0	....	....	16.4	83.6	82.2	3,718	1.45	35.8	82.0	+0.8	"	VI.			
89a	....	"	....	....	....	....	....	....	....	....	....	....	1.37	....	83.4	−0.6					
90	Cornish ...	"	62.4	20.3	....	4.0	....	.9	12.4	82.7	77.0	5,130	1.42	26.5	82.4	−5.4	Stein coal.....	VII.			
90a	....	"	....	....	....	....	....	....	....	....	....	....	1.41	....	82.7	−5.7					

TABLE I.—*Continued.*

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
			Heat Balance.							Efficiencies									
Reference Number.	Type of Boiler.	Type of Furnace.	Steam.	Hot Gas.	CO.	Ashes.	H_2O.	Radiation.	Unaccounted for.	Of Combustion.	Of Transmission and Utilization.	B. T. U. Transmitted p. Sq. Ft. of Surface per Hour.	Lbs. Gas per 1,000 B. T. U. Heat Gen. in Furn.	Lbs. Coal per Sq. Ft. of Grate per Hour.	Efficiency of Transm'n Calcul'd fr. Formula.	Difference between Observed and Calculated Values.	Coal Used.	Number in Original Report.	Reference where Complete Data may be Found.
91	Cornish ...	S'r (Schragr't)	30.0	20.5	....	3.3	....	1.4	5.8	80.5	38.9	5,180	1.89	32.5	71.1	−2.2	Stein coal.....	VIII.	
91a	.....	"	....	....	....	....	....	....	....	....	....	.....	1.83	....	73.7	−3.8			
92	Horizontal ret'n tubul'r	Stoker (Helix)	38.8	20.7	....	3.7	....	.8	6.0	80.5	77.7	2,847	2.30	16.1	69.5	+8.2	"	X.	
92a			....	....	....	....	....	....	....	....	....	.....	1.35	....	86.4	−8.7			
93	Combined .	(Ord'y) Carlo.	60.9	19.8	....	2.5	....	.7	16.2	79.7	80.1	1,877	2.40	15.6	72.4	+7.7	Washed coal...	XI.	
93a	.....	"	....	....	....	....	....	....	....	....	....	.....	2.02	....	78.6	+1.5			
94	Combined .	"	67.8	16.0	....	1.3	....	1.0	13.7	83.8	82.7	2,609	1.63	17.6	82.4	+0.3	" ...	XII.	
94a	.....	"	....	....	....	....	....	....	....	....	....	.....	1.38	....	86.1	−3.4			
95	Galloway ..	St'k'r (Leach)	66.3	24.0	....	.6	....	.7	8.4	90.3	75.6	5,707	1.57	18.9	75.6	0	Stein coal.....	XIII.	
95a	.....	"	....	....	....	....	....	....	....	....	....	.....	1.32	....	83.8	−8.2			
96	Galloway ..	"	69.1	19.4	....	.7	....	.7	10.1	88.5	80.0	5,868	1.40	18.6	79.6	+ .4	"	XIV.	
96a	.....	"	....	....	....	....	....	....	....	....	....	.....	1.13	...	85.4	−5.4			
97	Lancashire.	Generator....	57.4	24.1	....	10.9	....	3.2	4.4	81.5	72.5	5,019	1.84	24.3	70.7	+1.8	Coal	XVII.	
97a	.....	Gas	....	....	....	....	....	....	....	....	....	.....	1.70	....	74.3	−1.8	Brown coal.		
98	Lancashire.	Ordinary.....	60.1	23.0	....	5.2	....	.1	2.6	92.1	73.8	4,298	2.02	9.2	68.5	+5.3	Coke.....	XVIII.	
98a	.....	"	....	....	....	....	....	....	....	....	....	.....	1.93	..	70.8	+3.0			
99	Lancashire.	"	74.2	14.9	....	.9	....	1.1	8.9	87.1	83.1	4,608	1.53	7.4	78.7	+4.4	"	XIX.	
99a	.....	"	....	....	....	....	....	....	....	....	....	.....	1.11	....	86.9	−3.8			
100	C'mbined	Str. (Schulyrober) ...	55.6	25.4	.. .	2.7	....	.4	15.9	81.0	71.9	4,043	1.99	13.5	69.7	+2.2	Stein and brown	XX.	
100a	Cylinder .	Stoker (Schragrost)	....	....	....	....	....	....	....	....	....	.....	1.91	....	71.4	+0.5	mixed		
101	"	"	54.9	28.2	....	1.6	....	3.1	12.2	83.1	65.5	4,517	2.28	15.4	59.4	+6.1	"	XXI.	
101a	.....	"	....	....	....	....	....	....	....	....	....	.....	1.70	....	75.3	−9.8			
102	Cylinder ...	"	62.0	20.6	....	1.5	....	2.9	13.0	82.6	76.4	4,601	1.68	10.7	76.0	+0.4	Stein	XXII.	
102a	.....	"	....	....	....	....	....	....	....	....	....	.....	1.70	....	...	−0.8			
103	Lancashire.	Scheiber (ordinary)	35.2	22.0	....	.5	....	3.1	12.3	87.2	77.5	4,004	1.62	23.3	77.6	−0.1	Brown	XXIII.	

Bericht über Rauchfreie Dampfkesselanalegen in Sachsen, I. L. Lewicke, 1896

TABLE II.

UNACCOUNTED FOR HEAT AND RATE OF EVAPORATION.

B. T. U. Transmitted Per Sq. Ft. of Heating Surface Per Hour.	Less Than 2,500.	2,500 to 3,500.	3,500 to 4,500.	4,500 to 5,500.	Over 5,500.	All Tests.
Number of tests..	12	24	25	24	18	103
Total per cent. unaccounted for...	113.3	239.7	228.3	223.2	179.8	984.3
Average per cent. unaccounted for.	9.5	10.0	9.1	9.30	10.0	9.6

TABLE III.

UNACCOUNTED FOR HEAT AND RATE OF COMBUSTION.

Pounds Coal Per Sq. Ft. of Grate Per Hour.	Less Than 15.	15 to 25.	25 to 35.	Over 35.	Coal-dust Firing.	Miscellaneous.	All Tests.
Number of tests.	17	43	18	5	19	1	103
Total per cent. unaccounted for.	162.1	393.7	145.3	60.7	207.1	15.4	984.3
Average per cent. unaccounted for.	9.5	9.2	8.1	12.1	10.9	15.4	9.6

TABLE IV.

UNACCOUNTED FOR HEAT AND RATE OF EVAPORATION.

LANCASHIRE, CORNISH, AND GALLOWAY BOILERS ONLY.

B. T. U. Transmitted Per Sq. Ft. of Heating Surface Per Hour.	Less Than. 2,500.	2,500 to 3,500.	3,500 to 4,500.	4,500 to 5,500.	Over 5,500.	All Tests of These Boilers.
Number of tests..	1	4	11	17	13	46
Total per cent. unaccounted for...	5.8	53.5	101.2	150.7	151.9	463.1
Average per cent. unaccounted for.	5.8	13.4	9.2	8.9	11.7	10.1

TABLE V.

UNACCOUNTED FOR HEAT AND AIR SUPPLY.

Pounds Gas per 1,000 B. T. U. Generated.	Under 1.3.	1.3 to 1.6.	1.6 to 1.9.	Over 1.9.	All Tests.
Number of tests.....	23	37	20	27	103
Total per cent. unaccounted for....	199.2	320.5	186.3	278.3	984.3
Average per cent. unaccounted for.	8.7	9.7	9.3	10.3	9.6

In Table V. the quantity of gas is that in the flue, and includes all tests.

TABLE VI.

UNACCOUNTED FOR HEAT AND AIR SUPPLY.

Pounds Gas per 1,000 B. T. U. Generated.	Under 1.3.	1.3 to 1.6.	1.6 to 1.9.	Over 19.	All Tests.
Number of tests, no air leak.....	29	33	13	9	84
Total per cent. unaccounted for ...	262.1	327.2	117.4	92.0	798.7
Average per cent. unaccounted for.	9.1	9.9	9.0	10.2	9.5

TABLE VII.

CARBON MONOXIDE LOSS AND AIR SUPPLY.

Pounds Gas per 1,000 B. T. U. Generated.	Under 1.3.	1.3 to 1.6.	1.6 to 1.9.	1.9 and Over.
Number of tests, no air leak.....	29	33	13	9
Total per cent. of CO loss.....	19.5	43.2	3.0	1.1
Average per cent. of CO loss..... ...	.7	1.3	.2	.1

TABLE VIII.

UNACCOUNTED FOR LOSS AND TYPE OF FURNACE.

	Ordinary Hand firing.	Coking Stokers.	Coal-dust Firing.	Down-draft Grates.	Miscellaneous.	All Tests.
Number of tests.....	52	26	19	4	2	103
Total per cent. unaccounted for..... ..	475.5	256.1	207.1	22.6	23.0	984.3
Average per cent. unaccounted for.....	9.1	9.9	10.9	5.7	11.5	9.6

TABLE IX.

UNACCOUNTED FOR LOSS AND KIND OF COAL.

Coal.	Coke.	Anthracite.	Welsh.	Cumberland, Pocahontas and Clearfield.	Ruhr Coal.	Oberschlesische.	Scotch and Yorkshire.	Brown.	Mixed and Miscellaneous.	
Number of tests...	3	1	16	15	7	21	47	10	26	103
Total per cent. unaccounted for...	17.1	15.4	162.2	104.9	55.2	187.4	39.8	109.2	293.5	984.3
Average per cent. unaccounted for.	5.7	15.4	10.1	7.0	7.9	8.9	10.0	10.9	11.2	9.6

TABLE X.

Unaccounted for Loss and Type of Boiler.

	Cornish, Lancashire, Galloway.	Cornish, Multitubular.	Locomotive.	Vertical Tubular.	Combined.	Marine.	Babcock & Wilcox Type.	Heine Type.	Cahall Type.	Elephant.	Cylinder.	Horizontal Tubular.	Thorneycroft.	Gas Fired, Lancashire.	All.
Number of tests.....	45	2	6	3	8	5	4	9	2	2	4	8	4	1	103
Total per cent. unaccounted for.....	455.5	30.9	66.2	38.9	85.6	43.8	31.7	61.9	11.0	18.3	62.9	58.3	11.7	7.6	984.3
Average per cent. unaccounted for.....	10.1	15.5	11.0	13.0	10.7	8.8	7.9	6.9	5.5	9.2	15.7	7.3	2.9	7.6	9.6

TABLE XI.

Variations from Efficiency by Formula and Rate of Evaporation.

Double Tests on Shell Boilers (Lancashire, Etc.).

B. T. U. Transmitted per Square Foot per Hour.	Under 2,500.	2,500 to 3,500.	3,500 to 4,500.	4,500 to 5,500.	Over 5,500.	
Number of tests	0	2	10	18	10	20
Per cent. above formula..	.....	6.0	29.2	89.1	45.7	170.0
Per cent. below formula..	.....	10.7	17.7	73.5	37.4	139.3
Difference	.....	4.7	11.5	15.6	8.3	30.7
Average per test.....	.....	− 2.3	+ 1.1	+ .9	+ .8	+ .8

Double Tests on Water-tube Boilers.

Number of tests.....	2	4	1	.....	.....	7
Per cent. above formula..	0	20.7	7.0	.....	.....	27.7
Per cent. below formula..	28.8	20.0	5.4	.....	.....	56.2
Difference	28.8	1.3	1.6	.....	.....	28.5
Average per test	−14.4	−.3	+1.6	.....	.....	−4.1

Tests of Fire-tube Boilers (No Air Leak or Double Test).

Number of tests.....	6	11	10	2	4	33
Per cent. above formula..	11.2	31.5	28.4	5.5	4.3	80.9
Per cent. below formula..	14.3	14.5	5.6	5.5	.8	40.7
Difference	2.1	17.0	22.8	0	3.5	40.2
Average per test	−.3	+ 1.5	+ 2.3	+0	+.9	+ 1.2

Tests of Thorneycroft Water-tube Boilers (No Air Leak).

Number of tests.....	1	.....	1	1	1	4
Average.....	+1.2	.....	− 2.9	+ 1.9	0	+ .1

TABLE XII.

SAME AS TABLE XI.

B. T. U. Transmitted per Square Foot of Heat'g Surface per Hr.	Less than 2,500.	2,500 to 3,500.	3,500 to 4,500.	4,500 to 5,500.	Over 5,500.	All Tests.
Number of double tests	4	10	12	19	10	55
Actual efficiencies compared with computed :						
Per cent. above	9.2	43.4	32.7	96.1	45.7	22.1
Per cent. below	35.6	44.8	18.3	78.9	37.7	21.3
Difference	26.4	1.4	14.4	17.2	8.0	11.8
Average per test	− 6.6	− .1	+ 1.2	+ .9	+ .8	+ .2
Number of tests with no air leak	5	7	9	3	5	29
Per cent. above	3.2	14.8	24.9	7 4	4.3	54.6
Per cent. below	7.5	2.4	7.8	5.5	.8	24.0
Difference	4.3	12.4	17.1	1.9	3.5	30.6
Average per test	− .8	+ 1.8	+ 1.9	+ .6	+ .7	+ 57
Number of tests with air leak	3	7	4	2	3	19
Per cent. above	10.7	25.0	10.4	0	25.8	71.9
Per cent. below	1.8	1.6	45	6.4	4.1	18.4
Difference	8.9	23.4	5.9	6.4	21 7	53.5
Average per test	+ 3.0	+ 3.3	+ 1.5	− 3.2	7.2	+ 2.8

TABLE XIII.

VARIATIONS FROM EFFICIENCY BY FORMULA COMPARED WITH AIR SUPPLY.

Pounds Gas per 1,000 B. T. U. Generated.	Under 1.3.	1.3 to 1.6		1.6 to 1.9.	All Tests.
Number of double tests	14	15	16	10	55
Per cent. above	15.1	35.3	71.6	105.1	227 1
Per cent. below	46.7	63.0	73.1	32.2	215.0
Difference	31.6	27.7	1.5	71.9	12.1
Average per test	− 2.2	− 1.9	− .1	+ 7.2	+ .2
Number of tests no air leak	8	13	3	5	29
Per cent. above	4.0	29.2	7.9	13.5	54.6
Per cent. below	10.7	6.5	1.1	5.7	24.0
Difference	6.7	22.7	6.8	7.8	30.6
Average per test	− .8	+ 1.7	+ 2.3	+ 1.6	+ 1.1
Number of tests air leak	4	5	4	6	19
Per cent. above	.4	3.7	149	529	719
Per cent. below	8.5	8.2	17	0	184
Difference	8.1	4.5	132	529	535
Average per test	− 2.0	− .9	+ 3.3	+ 8.8	+ 2.8

TABLE XIV.

SAME AS TABLE XIII.

Unaccounted for Per Cent.	0 to 5.	5 to 10.	10 to 15.	Over 15.	All Tests.
Number of tests no air leak.....	6	12	9	2	29
Per cent. above	3.8	34.5	15.0	1.3	54.6
Per cent. below	3.6	8.1	12.5	.2	24.0
Difference	.2	26.4	2.5	1.1	30.6
Average per test.....	0	+ 2.2	+ .3	+ .6	+ 1.1
Number of double tests	3	26	17	9	55
Per cent. above formula	13.0	130.7	54.1	29.3	217.1
Per cent. below formula.....	0	108.1	81.4	25.5	215.0
Difference	13.0	22.6	27.3	38	2.1
Average per test	4.3	+ .8	− 1.6	+ .4	0

TABLE XV.

AIR SUPPLY ON DIFFERENT TYPES OF FURNACE.

GRATE :	Ordinary.	Stokers.	Down-draft.	Coal-dust.	Miscella-neous.	All.
Pounds gas per 1,000 B. T. U. in no air leak tests or with gas measured at bridge	40	23	2	18	1	84
Total	61.56	35.63	2.69	21.98	1.70	123.56
Average per test.....	1.54	1.55	1.35	1.22	1.70	1.47

TABLE XVI.

AIR SUPPLY AT DIFFERENT RATES OF COMBUSTION.

Pounds Coal per Square Foot of Grate per Hour.	0 to 15.	15 to 25.	25 to 35.	35 and Over.	Coal Dust.	Miscel-laneous.	All Tests.
Number of tests with no air leak or gas measured at bridge..... ..	12	32	16	5	18	1	84
Gas per 1,000 B. T. U.....	22.07	47.53	23.78	6.52	21.98	1.73	123.58
Average per test..... ..	1.84	1.49	1.49	1.30	1.22	1.73	1.47

TABLE XVII.

AIR SUPPLY WITH DIFFERENT KINDS OF COAL.

COAL:	Coke.	Welsh.	Cumberland, Pocahontas, Clearfield.	Ruhr.	Upper Silesian.	Scotch and Yorks.	Brown.	Miscellaneous.	All.
Number of tests (no air leak or gas measured at bridge).....	2	8	9	7	21	4	9	24	84
Pounds gas per 1,000 B. T. U..	3.05	12.99	13.7	9.53	30.16	6.85	13.57	33.73	125.38
Average.....	1.55	1.62	1.52	1.36	1.44	1.71	1.51	1.41	1.47

DISCUSSION.

Mr. William Kent.—If any other student of the subject of boilers had taken these 100 tests and tabulated them as Mr. Hale has done, and studied them throughout, I think he would probably reach the conclusion which Mr. Hale has practically reached, but which he does not exactly state in words, namely, that the study of 100 tests of different coal and different boilers, made by different people under all sorts of conditions, will not enable us to say what are the laws that govern the results of these tests. I think that conclusion is about what Mr. Hale has got, and it would appear therefore that the whole study is love's labor lost.

1. The definitions of three kinds of efficiency, credited to Hudson, seem to be applicable strictly only to internally fired boilers of the marine or locomotive type. In externally fired boilers the radiation loss is of two kinds. (*a*) That which passes through the brick walls and never enters the boiler; (*b*) that which is radiated from the steam space in the boiler through the metal and its outer covering. For such boilers a revision of the definition seems necessary.

2. The author takes the efficiency of combustion as the sum of the heat utilized in making steam and the loss of heat in the chimney gases, neglecting radiation. It does not seem right that this should be neglected. Where it is not known it would seem better to assume it at 2 per cent. or 3 per cent. than to assume that it does not exist at all.

3. The author says that "the higher the rate of combustion the

less the unaccounted for loss up to the limit of 35 pounds per square foot of grate per hour. The high loss above this limit is probably due to particles of coal carried up the chimney by the strong draft." I see no reason for believing that the increase of 4 per cent. in the loss unaccounted for can be thus explained. It is due, no doubt, to the fact that all the tests above the limit of 35 pounds were of brown coal, except one. In that one the rate of driving was the highest in the whole list, test No. 4, Thorneycroft boiler, 66.8 pounds per square foot of grate per hour, but the unaccounted for loss is put down as only 2.3 to 3.9 per cent. The only conclusion that I can draw from Tables II., III., and IV. is that the loss unaccounted for is independent of the rate of combustion or of the rate of driving of the boiler.

4. From Tables V., VI. and VII., I would draw the conclusion that there is no relation between the loss unaccounted for and the pounds of gas per 1,000 B. T. U. generated.

5. From Table VIII. I would draw the conclusion that the loss unaccounted for is practically independent of the method of firing, except that the down-draft furnace appears to greatly decrease this loss, if four tests are enough to judge from.

6. From Table IX. I would draw the conclusion that there must be some mistake in the reported results of the test with anthracite coal, No. 80, either in the reported evaporation, in the reported heating value of the coal, or in the gas analysis from which the reported loss of heat in the chimney gases is computed. That the loss of heat in the gases could be only 10.2 per cent., and at the same time the unaccounted for loss could be 15.4 per cent., is evidence of some error. In the same table the loss unaccounted for in the brown-coal tests is given as averaging 10.9 per cent., but if we omit the dust-firing tests with this coal, in which the conditions of combustion are very different from those with other methods of firing, we find that the loss unaccounted for ranges from 12.0 to 16.4 per cent., averaging 13.6 per cent. in six tests. The loss unaccounted for is therefore very much greater with the brown coals, dust-firing tests excepted, than with any other coals.

7. The author says: "The horizontal return tubular boilers appear, all things considered, to be the type of boiler for best economy of steam production." I can find nothing in the paper which supports this conclusion. The figures of the heat balance in Table I. and those of the loss unaccounted for in Table X. do not appear to support it. It is not to be expected that any satis-

factory conclusion can be drawn on this point from such a heterogeneous lot of tests as those given in the paper, with boilers of all sorts of proportions, tested at different places, with different fuels and furnaces, and with different rates of driving. It is especially difficult to draw conclusions when we have no means of knowing whether the heating values of the coal or the gas analyses are accurate. This same criticism applies with equal force to nearly all the conclusions drawn in the paper.

8. The author says that the "hydrocarbon gases in the flue are the loss to which we can ascribe this 7 per cent." unaccounted for. The tables show numerous instances in which there appears to have been an abundant supply of air and no CO in the chimney gases. It is difficult to believe that in all these cases, especially with semi-bituminous coals, there can be any great amount of hydrocarbon gases unburned, yet we find some cases in which with these coals the unaccounted for loss is over 7 per cent. Test No. 4, made by Mr. Hale himself, is one of these. With Pocahontas coal, burned at 20 pounds per square foot per hour, with 1.54 pounds of gas per 1,000 British thermal units generated, indicating abundant air supply, and with no CO in the chimney gases, there is an apparent loss unaccounted for of 9 per cent.

There are two suppositions which may account for this loss unaccounted for (besides errors in the reported heating value of the coal, and errors in gas analyses, both of which are highly probable): 1. Firing large quantities of coal at a time, resulting in choking the air supply, and the sudden volatilization of large quantities of hydrocarbon gas which escape unburned. If a long time elapses after such firing the coal bed burns down, and the gas analyses, averaged during the whole time between consecutive firings, will show an abundant air supply, but will not reveal the fact that for two or three minutes during that time the air supply was deficient and the hydrocarbon gases were escaping unburned. 2. The generation of water gas from the decomposition of the moisture in the coal, and its escape unburned. This must necessarily take place if we have the conditions as follows: First, a thick bed of white-hot coke on the grates; second, a heavy firing of fine coal, containing a large quantity of moisture, such as is found in the coals of our Western States and in the brown coals and lignites. The fresh coal chokes the air supply, the moisture suddenly evaporates while in contact with hot coke, and the reaction $C+H_2O=CO+2H$ takes place. This $CO+2H$, or water

gas, finding no free oxygen present, escapes unburned. In a series of tests with Western coals which I made three years ago, taking gas samples one, two, three, and four minutes after firing, I often found that the first sample contained as high as 7 per cent. CO, and no free oxygen, and the second contained 5 CO and 1 or 2 O. In such cases the loss unaccounted for was from 15 to 23 per cent., the latter being a higher figure than any of those given by Mr. Hale. A careful study of these tests at the time led to the conclusion that the only way to account for this very high loss unaccounted for was the supposition that in addition to the rapid volatilization of the hydrocarbons in the absence of any free oxygen to burn them was the generation of water gas from the decomposition of the moisture in the coal, and I have as yet found no reason to believe that the supposition, which was original with myself, was not a correct one.

The remedy for this unaccounted for loss is clearly apparent. 1. The firing should be done so as to avoid the rapid volatilization of great quantities of gas and the evaporation of moisture at one time. 2. The gas must be brought immediately into intimate contact with very hot air. 3. There should be a fire-brick combustion chamber in which the complete combustion of the gas may take place before it is allowed to come in contact with the comparatively cool surface of the boiler. If this triple remedy were generally adopted we would not only save the unaccounted for loss, say 10 to 20 per cent. with Western coals, but would abolish the smoke nuisance, "a consummation devoutly to be desired."

One of the chief objects of the paper is the study of the "unaccounted for loss." I think that the author fails to get one of the reasons why there is this loss. He mentions one reason—the hydrocarbon loss, but he seems to consider that that is the only loss unaccounted for. He speaks of the investigation necessary in regard to air supply, but he does not seem to notice at all the fact that the air supply in ordinary furnaces fired with bituminous slack coal varies every second during the time from one firing to the next. If we are firing slack coal with some moisture in it, the air supply immediately after firing is practically nothing; we have choked off all the air supply; we have sudden volatilization of gases, which makes a partial plenum in the furnace, checking the draft, and we have distillation of water from the moist coal. At this instant our analysis shows no oxygen in the gases. For the first minute after firing bituminous coal, four shovelfuls on

each side the grate, I have found no oxygen in the chimney gases and as high as 7 per cent. carbonic oxide. If I had waited ten minutes before the next firing no doubt I would have an enormous supply of air. So the air supply is a thing which cannot be regulated at all by hand firing, except within such certain limits as are possible by repeated firings. It is possible with the use of mechanical stokers and other devices, with expert firing, to regulate the air supply pretty uniformly throughout the tests, but it is not possible with ordinary furnaces and ordinary firing, where soft coal is used. In a series of seventy-five tests which I made about three years ago I found the unaccounted for loss to range from 4 per cent. to 23 per cent. The ordinary way of accounting for it, namely, the distillation of hydrocarbons, would not account for the higher figure at all. So I framed a new theory which I submit to Mr. Hale for his consideration. It is that this very serious loss occurs with moist coal. When we fire on a bed of white-hot coke several shovelfuls of bituminous coal, high in moisture, we have all the conditions necessary for making water gas. Illinois coal oftentimes contains 14 per cent. of water. That water is almost immediately vaporized. An analysis of the chimney gases taken at this time shows no free oxygen. The water is decomposed, making water gas, carbonic oxide and hydrogen, by the reaction $C+H_2O=CO+2H$. We are turning the furnace into a hydrogen producer. And since there is no oxygen to burn that hydrogen, it, together with the carbonic oxide, escapes unconsumed, and that accounts for the great loss. That loss happens during the first half minute or minute after firing, and then ceases. That loss, together with the loss due to the great volatilization of the extremely volatile matter of these Western coals, will account probably for this loss, even to the 23 per cent. The high figures of loss unaccounted for occur when the coals contain a great quantity of very easily volatilized matter together with a great deal of moisture.

Mr. Hale speaks on page 779 of the phenomenon of dissociation. I think there are no experiments on boilers in which it has been shown that such phenomenon occurs at all. If it did occur the gases thus dissociated would combine either at the bridge wall or beyond, so we would lose nothing. I think there is no evidence that such dissociation takes place.

On the last page the author says: "The results show that present knowledge enables the boiler efficiency to be predicted

reasonably well, provided the efficiency of combustion and the air supply are accurately known." That is, we can predict with reasonable certainty the efficiency of a boiler provided we know things which are impossible to be known. He says further: "The rate of burning per square foot of grate affects the air supply." I don't know where he gets that idea. The rate of burning does not affect the air supply, although the air supply does affect the rate of burning. We can burn a hundred pounds per hour and get the same percentage excess of air supply per square foot of grate that we can with any other rate of burning. The excess air percentage is simply a question of the thickness of the fire and the mechanical character of the coal, as to how it obstructs the fire, and of the relation of these factors to the draft. If we put draft enough on a thick bed of fire you can get a tremendous access of air, or, with not draft enough, a deficiency of air.

Mr. Hale says: "The most pressing need, therefore, in the study of boiler efficiency is the further determination of the factors that govern the efficiency of combustion and the air supply." The efficiency of combustion is secured by intimate mixture of hot air with the gases that proceed from the coal at the proper time and in the proper place, surrounded by firebrick, and that is accomplished in certain ways by the down draft of the furnace, by certain kinds of automatic strokers, by certain kinds of furnaces and methods of firing, all of which Mr. Hale does not go into. I agree with him entirely that a few more accurate tests, in which all the quantities, such as thickness of fires, gas analysis, etc., are observed, are, for the purpose of increasing our knowledge of boiler efficiency, worth these hundred tests in which part of those things are mentioned.

Colonel Meier.—There is so much matter in this paper which has evidently been very carefully collected that I would not like to attempt a criticism of the whole of it. I can only take up a few points which appeal to me directly. For instance, in the general comparison between the types of boilers I note that Mr. Hale quotes 40 tests on internally fired boilers; 33 on externally fired tube boilers; only 7 on water-tube boilers, and 4 on Thorneycroft. I think the number of tests on water-tube boilers and on the Thorneycroft, compared with the total number, is altogether too small to draw such a sweeping conclusion as he does at the end of his paper, where he states that horizontal tubular boilers are the best for economy, and then puts the Thorneycroft result as

the best of all. Now it is easily seen that when you have only four tests you may have got the very best tests of those and you may also have the worst. In the same way, when you have only seven tests of the water-tube boilers you may have the worst or the best. Especially in selecting tests, as Mr. Hale has done, mainly because they contained all the data he needed for his illustrations, I think that conclusion is made on too small a number of results, and I do not think it is fair, and it is entirely in the face of the general tendency in this country. In fact, we all know that water-tube boilers have won the field almost entirely now. And if one may be allowed to speak in matters in which one is particularly interested, I would just call attention here to the fact that he gives only two tests of Heine boilers, and those are foreign tests. I will only note one thing, that the number of heat units transmitted per square foot of heating surface is about one-half—yes, a little less than one-half—of our standard practice in this country; that therefore it does not justly represent the American Heine type. Furthermore, I know nothing of the particular coal. I would have to study into that more fully in order to criticise it; for the nature of the coal is an essential fact in the problem. But I do state very positively that when we run at so low a rate of transmission of heat per square foot of heating surface as given there, we invariably get poor results; that we need to run about double as much. We have got to do double as much in quantity before we get those results in quality of which the boiler is capable, and to which it is fully entitled in such a comparison.

*Mr. R. S. Hale.**—Mr. Kent in his discussion repeats a number of the conclusions that I have reached, especially the fact, which I originally stated on page 774, that many of the results must only be considered as indications to be confirmed or reversed by future experiment. One or two of his statements, however, should be briefly replied to.

1. In regard to the definitions of the three kinds of efficiency, Mr. Kent's criticism to the effect that the radiation from the furnace should be considered differently from the radiation from the boiler walls is theoretically correct, but since even the whole radiation appears to be such a small proportion of the heat loss the theoretical error is unimportant.

2. In regard to column 11 in Table I., giving the efficiency of combustion, the text of the paper states that this is in error by

* Author's closure, under the Rules.

the amount of the radiation, but Mr. Kent's suggestion, if it had been adopted, would have made the error even greater, since Mr. Kent would like me to assume radiation at 2 or 3 per cent., whereas, as a matter of fact, the radiation expressed as a percentage is very variable, but is not much over 1 per cent. on an average. Column 11 does not assume that the radiation does not exist, as may be seen on referring to the text.

3. The statement on page 775 in regard to the excess of unaccounted for heat when the rate of combustion is above 35 pounds of coal per square foot of grate is not exactly contradicted by Mr. Kent, but he offers a further suggestion for accounting for the variation in results above this rate of combustion which appears to be also very probably correct. The essential point is that this particular figure is an accidental variation, and that the unaccounted for heat decreases as the rate of combustion and temperature of the fire increase. In his paragraphs 4, 5, and 6 Mr. Kent restates some of my conclusions in different words, while paragraph 7 merely gives in different words what I stated at the bottom of page 774. In paragraph 8 Mr. Kent expresses a disbelief that hydrocarbon gases can be given off from the semi-bituminous coals in sufficient quantity to cause a loss of unaccounted for heat of 7 per cent., although $\frac{1}{10}$ per cent. by volume of some of these gases would be sufficient to account for this loss. In regard to Mr. Kent's suggestion that water gas from the decomposition of the moisture in the coal may account for the unaccounted for loss, I may point out that the formation of water gas takes place when H_2O in the gaseous condition is passed over red-hot carbon, whereas, even when moist coal is fired on top of the bed of red-hot coals, H_2O in a solid condition is merely placed in contact with the red-hot carbon. This gives very slight opportunity for the formation of water gas, since even if the H_2O evaporates it goes upwards through the fresh coal and not downwards through the red-hot coal. I may further note that if the formation of water gas took place in this way there would be very much greater loss when moist coal was used than when dry coal was used, yet as a matter of fact there is no evidence that wetting coal before firing it reduces the efficiency by more than the loss of heat due to the evaporation of this moisture. Still further, I may note that the gas analysis with the Orsat-apparatus furnishes a ready method for the detection of CO, so that if the unaccounted for loss was due to water gas, then whenever this unaccounted

for loss was high the amount of CO present in the gas should also be high, yet the tests give not the slightest indication of this occurring.

Mr. Kent's suggestions as to the methods of firing best adapted to avoid the unaccounted for loss may be very good, but unfortunately are not accompanied by any evidence.

In regard to Mr. Kent's statement that the air supply cannot be regulated by hand firing, I can only suggest that he do some experimenting on this subject. In my tests I have certainly found it possible to regulate the air supply to a very great extent by my directions to the fireman. Of course it is not possible to keep the air supply absolutely constant with hand firing; neither is it possible to do this with stokers or down-draft furnaces.

Mr. Kent says that there are no boiler experiments which show that dissociation occurs. This is very natural, as to perform such an experiment in a boiler furnace would be rather difficult. Le Chatelier's experiments, however, which may be found reported in the *Zeitschrift für Phys. Chemie* in 1888, page 784, show that dissociation of CO_2 occurs at about 1,500 degrees centigrade, a temperature which certainly occurs in a bed of coals. Berthelot's and Deville's experiments, reported in the *Comptes Rendus*, show the same thing. It is not necessary, however, to assume that dissociation takes place, since undoubtedly the radiation from the beds of coals reduces their temperature at low rates of combustion.

Mr. Kent says that he does not know where I got the idea that the rate of burning per square foot of grate affects the air supply. I can only refer him to the results given in the paper. It is undoubtedly true that the rate of combustion is not the only thing that affects the air supply. Probably it is the relation between the rate of combustion and the square root of the draft pressure, while it is also affected by the thickness of the fire and the mechanical character of the coal.

In reply to Colonel Meier, I can only agree with him that the number of tests is too small to draw any absolute conclusions; but, as stated in the text, many of the conclusions are only to be regarded as indications to be confirmed or reversed by future experiments. I took all the tests that I could find in American and foreign literature which give the sufficient data; and if only two of these were on the American Heine type of boiler, I can only regret that this type has not had more thorough tests in the

way necessary for a complete study of its efficiency. The American Heine type has had a fine commercial success, and I wish that Colonel Meier or his associates would furnish tests and data from which it might fairly be given its place among other boilers in regard to its economy.

DCCCXX.*

ELEVATORS.

BY CHARLES R. PRATT, MONTCLAIR, N. J.

(Member of the Society.)

THERE is little to interest the engineer in the early history of elevators. Vitruvius describes an elevator built by Archimedes (236 B.C.), operated by man power applied to a capstan revolving a drum on which the hoisting ropes were wound. Very little advance in the art was made from that time until George H. Fox & Co., of Boston, built a worm-gear elevator, in 1850. That this vertical form of railway did not follow the advance of horizontal railways is of course due to lack of demand for that class of transportation during the early application of steam power. Localization of commerce has filled its great centres of distribution with merchants and merchandise to an extent which has added story upon story to our buildings, until primitive hoisting apparatus and stairways can no longer accommodate this vertical traffic.

Means for raising freight or passengers have always been adequate to the demands of the times, and the elevator engineer has never hesitated for an instant to furnish greater speed, travel, or lifting capacity when called upon to do so. Mechanical connection with line shafting, direct-connected steam hoisting engines, hydraulic or electric hoisting machines operated by an isolated plant or from an outside source of power, are all capable of operating elevator cars at any desired speed or load with perfect safety and comfort.

There is, however, so much elaborate and interesting detail in an elevator system, and such a variety of types, as to merit the consideration of this Society, especially as there is very little literature—aside, alas! from a voluminous amount of vituperous and unprofitable discussions—upon this subject. Let me state

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

at the beginning that no treatise on elevators can be written in the space permitted in a paper before this Society; there are too many radically different types to sketch more than briefly the salient features of the most important; and the writer trusts that these will interest the Society to the extent of bringing out more complete papers upon individual types.

Let us consider first, briefly, what the requirements of an elevator are, and then describe the different means used to meet them. In order of importance they are:

1. Safety.
2. Reliability.
3. Durability.
4. Economy.
5. Control.
6. Comfort.
7. Speed, load, and travel.
8. Compactness.

Safety.

Let us classify this by the things which are unsafe for an elevator to do:

1. To fall unretarded to the bottom of the hoistway.

2. To be thrown by its counterbalance up against the top of the hoistway.

3. To be stopped at too great a speed in its descent by a safety catch on the car which will not stop it gradually enough to avoid injury to the passengers, or by a safety catch stopping only one side of a car not built to stand a diagonal strain and thereby collapsing the car.

4. To pay out hoisting rope after the car has been stopped by anything except its own hoisting machine, in its descent.

5. To let the hoisting machine continue to hoist after the car has met the top of the hoistway, thereby breaking some connection between the hoisting machine and the car, and depending upon the car safeties to prevent the car from falling to the bottom of the hoistway.

6. To let the hoisting machine run the counterweights into the top of the hoistway and possibly drop them on top of the car.

7. To lose control of the hoisting machine in such a way as to alarm the passengers to the point of jumping on or off a mov-

ing car and getting caught between car and hoistway door sills and lintels.

These are some of the unsafe features of some elevators which have been in general use for the last forty years, and can be taken as the typical conditions for which adequate safety appliances are provided on all first-class elevators. There are two general methods used to prevent a car from falling unretarded to the bottom of the hoistway. One is to stop or retard the car before it attains undue speed, and the other allows it to fall unretarded till it reaches a certain distance from the bottom of the hoistway, where it is brought to a gradual and safe stop. This latter is only used in addition to and never as a substitute for the first method.

A description of all the devices to stop or retard an elevator car by this first method would nearly fill a year's *Transactions* of this Society. I will, therefore, describe only the typical ones in general use.

The most simple device is that operated by the breaking of the hoisting ropes, which, unfortunately for the utility of this device, is the rarest cause of a car falling. When, however, this does occur—and the break must be near enough to the car, if the hoisting machine is located in the lower part of the building, to prevent that part of the rope between the hoisting machine and the overhead sheave from keeping sufficient tension where it is fastened to the car to prevent the device from working—the rope breaking in such a way as to relieve it of all tension at the car, a safety catch is thrown into action, gripping or locking on some support in the hoistway.

This device is sometimes used in addition to more reliable means, but is not considered as at all necessary by the leading elevator companies. Its general design is as follows:

The hoisting ropes are either connected to the car by a system of levers, which are operated by a spring to throw the safety catches when the ropes slack, or they are connected to the car by levers or sheaves, which balance the tension of the hoisting ropes against the tension of the counterweight ropes, allowing the counterweight ropes to operate the safety.

This device is a very pretty exhibition safety. All that you have to do is to break the hoisting-rope connections at the car, and the car is instantly locked to its safety supports. It can also be operated by grasping the hoisting ropes from a landing

door above the car, drawing them towards you, and letting them snap back. This test always satisfies elevator inspectors that the elevator is perfectly safe.

All other means used to prevent an elevator car from falling unretarded to the bottom of the hoistway are operated by acceleration of car speed, which is the only reliable means of operating an elevator safety. Inertia, pneumatic and hydraulic resistance, and centrifugal force are thus used.

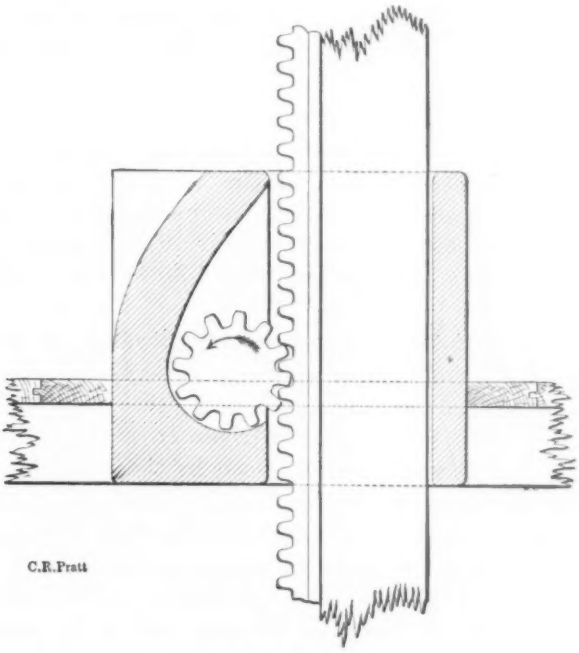


FIG. 272.

Inertia devices have only been tried on very slow-moving cars. One in the form of a sort of floating pinion held in a pocket on the side of the car and meshing in a rack secured to the hoistway, is shown in the accompanying sketch (Fig. 272).

As the car descends the pinion revolves, as shown by the arrow, and under a slow speed remains in its pocket, but any sudden acceleration rolls it up the incline of the pocket and jams it in the rack. Increased friction in the rack may have as much to do as inertia in this operation, but it certainly does its work all right on very slow-speed cars.

Another inertia device consists in heavy pendulums attached to the car and oscillated by waving cams running up each side of the hoistway. The length of the pendulums and the curves of the cams being determined to synchronize with the car speed, no retarding effect is obtained until the speed increases, and then there is trouble. The noise and vibration of a car having nothing but this device to sustain it against the force of gravity can better be imagined than described.

Pneumatic resistance is used to operate an elevator safety catch either by a light wood fanboard suspended below the car on balanced levers and filling the hoistway as much as possible, accelerations of car speed lifting it by pressure of the air and operating the safety catches, or else by a pneumatic piston running like a counterbalance in a closed box, the rope which carries it being connected to the car by a spring balance, causing it to operate the safety catches when the tension on its ropes is increased by increased air resistance due to acceleration. Neither of these pneumatic devices is considered to be reliable.

Hydraulic resistance to prevent an elevator car from falling unretarded to the bottom of the hoistway is applied simply as a hydraulic power pump driven by the car. It is usually located over the hoistway and driven by a rope from the car when the car descends, and reversed by the counterweight rope when the car ascends, having its circulating valve wide open during the ascent of the car. This device has all the attributes of perfect safety, but adds materially to the friction losses of the elevator and to the cost of installation and maintenance.

We now come to the most popular device for operating an elevator-car safety—a centrifugal governor. This is located either on the car, acting directly on the safety catches, or over the hoistway, where it operates the safety catches on the car by retarding the rope which is fastened to the car and drives the governor.

Having described all methods in general use for operating the safety catches, we will now consider the catches and their merits.

The rack and pawl is only adapted to slow-speed cars and where they are operated by the breaking of the hoisting ropes, since they could never stand the impact of a car falling at any speed above 100 feet per minute.

Another form of safety catch grips the smooth steel guide rail with as little slip as the rack and pawl, and is equally unfit to be operated by a centrifugal governor or any other device on a high-speed car. It has, however, been used to a great extent under such conditions. Its general design is shown in the following sketch (Fig. 273), where C is a sharply corrugated hard-steel roll, and is shown gripping the steel guide rail B by rolling up the incline of its pocket in the safety block A, which is attached to the car. It nominally rests in the bottom of this pocket clear of the rail B, and is lifted up into contact by a tripping device.

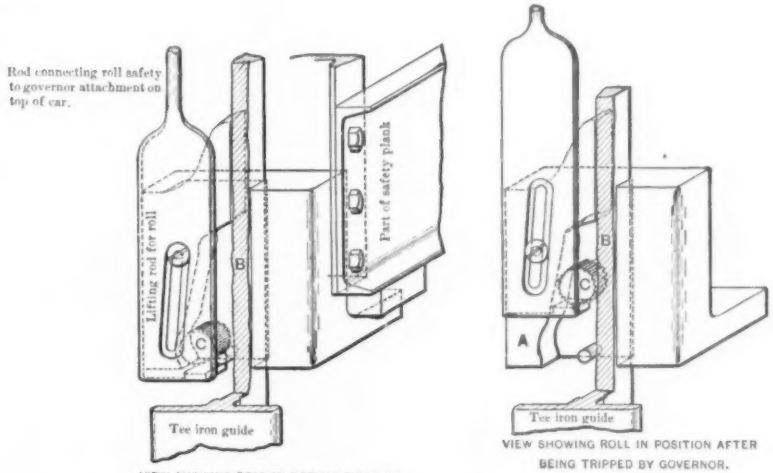


FIG. 273.

Another form of steel-rail grip is operated by the ropes leading from the hoisting machine, making several turns around drums between the top crossheads of the car, and leading up over sheaves and down again to the counterweight. The drums between the car crossheads are mounted on shafts which revolve in nuts fastened to the crossheads—a right-hand nut in one crosshead and a left-hand nut in the other; the ends of the crossheads bearing on either side of the steel guide rails act as braking clamps to check the car speed in case the ropes break between the car and the hoisting machine, thereby allowing the counterweight to overhaul these ropes, revolve the crosshead drums, and screw the crossheads together against the rails.

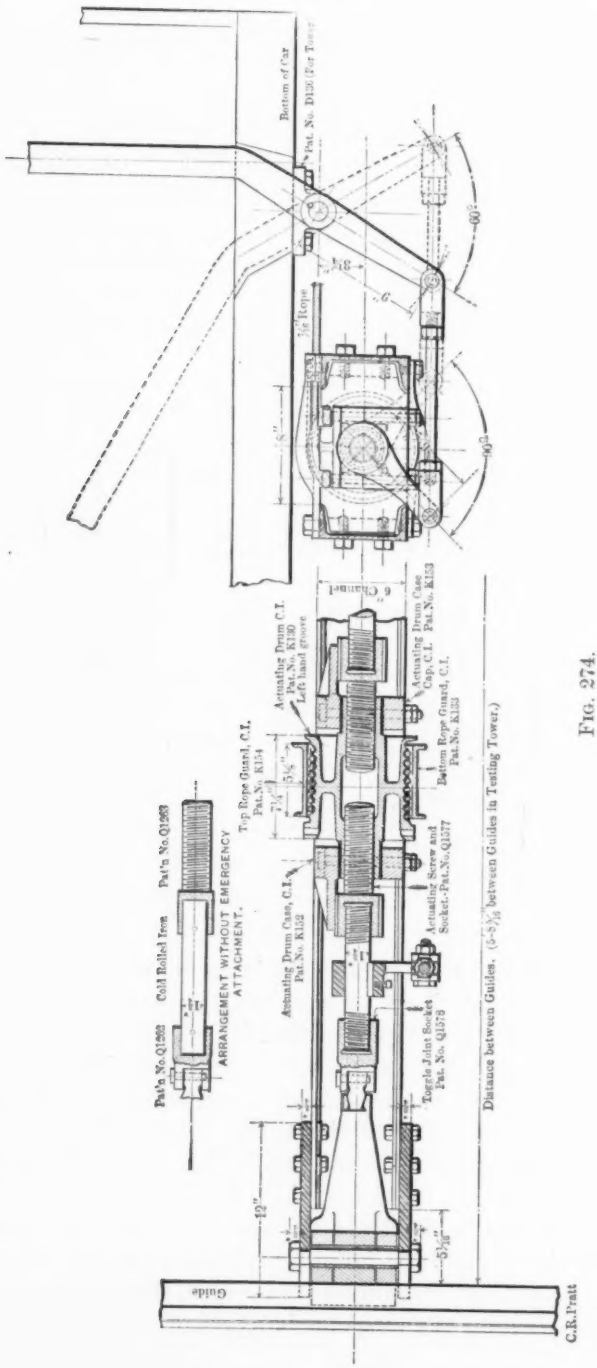

Bottom of Car
Pat. No. D130 (For Tower)
60°
60°
18"
6"
8"
⅝" Rope
6" Channel
Actuating Drum C.I.
Pat. No. K130
Left hand groove
Actuating Drum Case
Cap, C.I. Pat. No. K133
Bottom Rope Guard, C.I.
Pat. No. K133
Top Rope Guard, C.I.
Pat. No. K134
7¼"
5⅛"
Actuating Drum Case, C.I.
Pat. No. K132
Actuating Screw and
Socket-Pat. No. Q1077
Toggle Joint Socket
Pat. No. Q1078
Pat'n No. Q1262
Cold Rolled Iron
Pat'n No. Q1263
ARRANGEMENT WITHOUT EMERGENCY
ATTACHMENT.
12"
5⅛"
Distance between Guides. (6-5/16" between Guides in Testing Tower.)
Guide
C.R. Pratt

FIG. 274.

The limitations of this device are its doubtful gripping powers and the limited functions of its operating device; it could not be operated by a centrifugal governor with the rope connections here described.

Wood guide rails form the easiest means of bringing an elevator car to a safe and gradual stop when it is descending too rapidly, and there are many forms of gripping devices for this purpose. The sharp-toothed cam is all right, provided it is not designed to impinge the rail to such a depth as to stop the car too suddenly. Other wood-rail grips either cut deep shavings or crush the rail as they slide. Wood rails, however, are becoming less used every year, as they cost as much as a steel rail when they are constructed and erected in a first-class man-

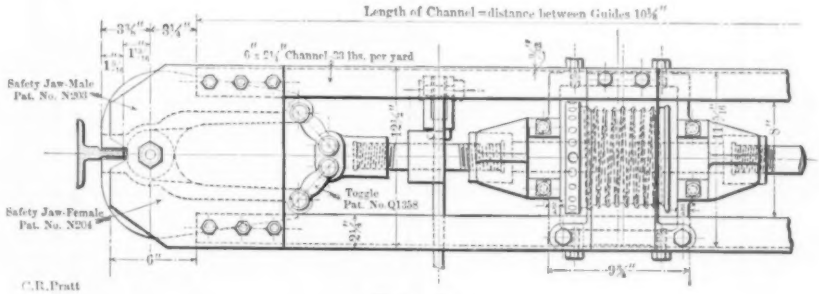


FIG. 275.

ner. Their present use is principally for cheaper grades of elevators and dumb-waiters.

The best steel-rail gripping device has vise-like jaws, which are supposed to slide far enough along the rails to stop the car easily. The first form of this device brought into use, aside from that which screws the crossheads together as before described, had jaws operated by toggle joints, actuated by a right and left-hand threaded screw revolved by a rope controlled by a centrifugal governor, as shown in the accompanying drawing (Fig. 274).

This device has been modified, as shown in the following drawing (Fig. 276), to use spring power to operate the jaws, instead of operating them by the governor rope direct.

A car safety of this same general character, designed by the writer and built in four sizes, varying from 2,500 to 40,000 pounds capacity, has stood some very satisfactory tests. Its construction is as follows:

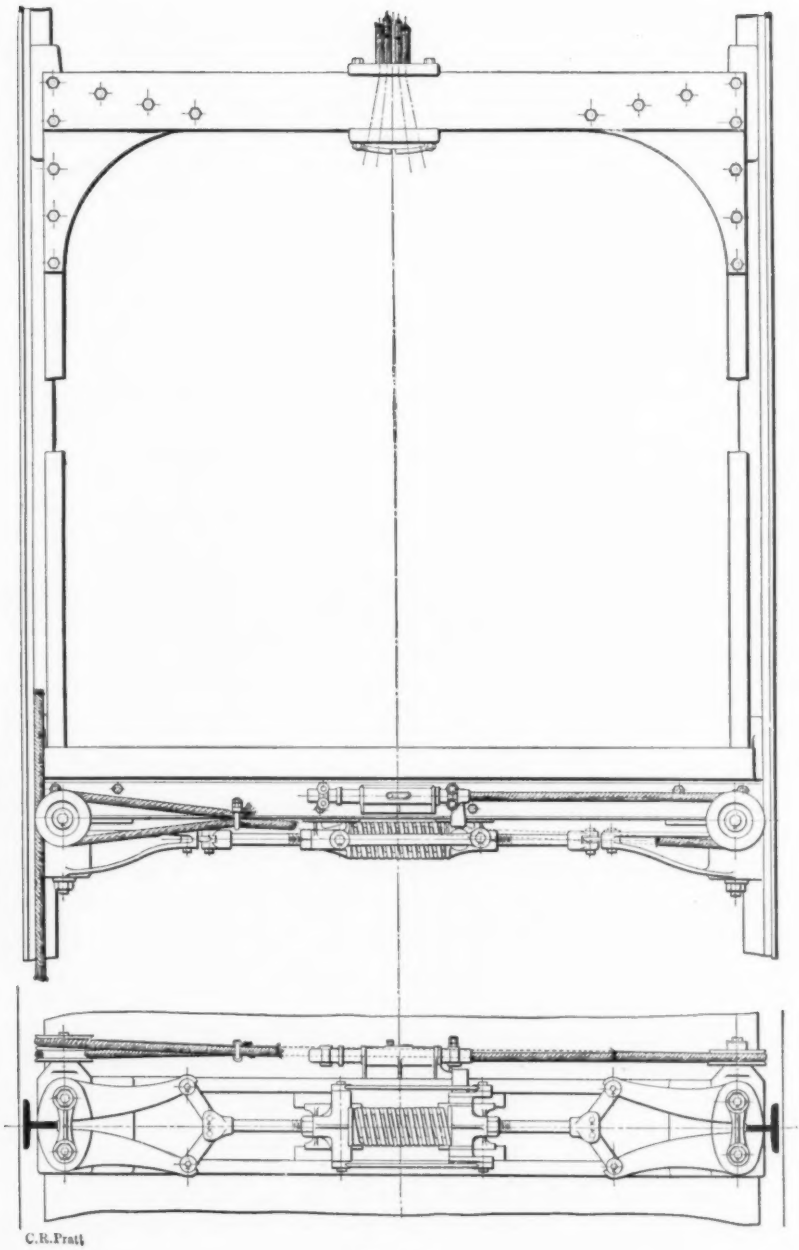


FIG. 276.

As shown on the plan, Fig. 278, the car guide rails *AA*, which are steel tees accurately machined and highly polished by the guide shoes of the car and heavily lubricated, are gripped on

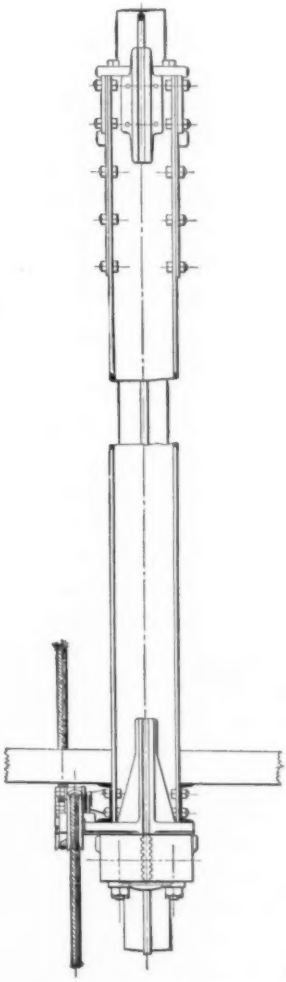


FIG. 277.

either side by the jaws *BB* and *B'B'*. These jaws are held on the ends of levers *CC* and *C'C'*, which are pivoted between jaw plates *D* and *D'* and guide-shoe stands *C* and *C'* by eccentric pins; the angular adjustment of these eccentric pins by means of

a locking lever forms a means of taking up the wear of the jaws by setting them nearer the rails.

On the other end of each jaw lever are two rolls, and between the rolls of each pair of levers are wedges which, by forcing the rolls apart, force the jaws on the rails. Wedges *EE* are formed on the ends of rods *FF*, which are tapped into springhead *F*. Wedges *GG* are bolted to eye on the end of rod *H*, which is tapped into springhead *H*, thus using the helical spring *J* under compression and drawing the two pairs of wedges together by its extension. We have here a device that applies a definite pressure to the rails, is adjusted for any pressure required to stop the car easily, and insures exactly the same pressure on each rail, relieving the car of diagonal strain.

For the benefit of any who may entertain that ancient prejudice against the durability of springs, I will state that these particular springs, after being held closed for over a year and worked their full working distance over 500 times, lose 3 per cent. of their free length and gain from 15 to 20 per cent. in their resistance to compression.

Fig. 279, the side elevation of this safety, shows the action of the centrifugal governor in releasing the spring and operating the wedges. The weights *KK*, pivoted on pins *LL*, are held against centrifugal force by a torsional spring wound on the governor sleeve and acting on the weights through the collar *M* and links *NN*, which keep the two weights at equal distance from the centre of the spindle. Tripping speed adjustment is obtained by regulating the torsional resistance of this spring by turning the split clamping collar *O*, to which one end of the spring is secured.

In hoisting, the governor weights may hit the trigger *P* without tripping it, but as centrifugal force increases their radius of rotation in lowering, until they reach the trigger, the first weight that touches it knocks it off at the first blow. The Pickering governor is very generally used to trip elevator safeties, but the writer's experience with it discovered that it sometimes tripped the safety when the car was being hoisted, and had a way of gently tapping the trigger almost off as the car descended, so that it would jar off at some unexpected moment when it was not wanted to do so.

The governor shown in Fig. 279 is revolved by the rope *S*, which is secured to the top of the hoistway and leads to the

FIG. 279.

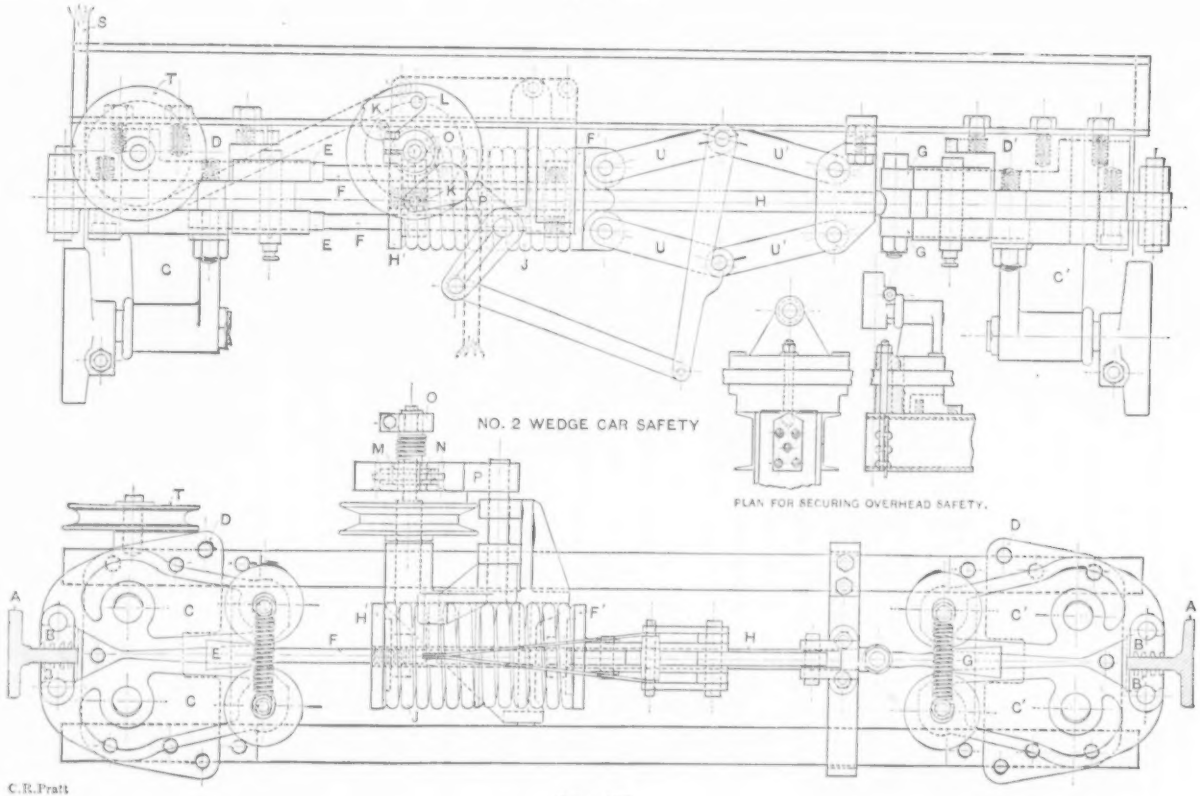


FIG. 278.

governor pulley over the idle sheave T, and is kept under proper tension by a weight at the lower end of it.

Means for releasing the safety from inside of the car is obtained by a screw acting on the toggles $U\,U$ and $U'\,U'$ by means of the nut V and links $W\,W$. This enables the operator to put his elevator in operation again without delay if the safety was only sprung by undue speed of the hoisting machine, or lower the car to the next landing and let the passengers out if any accident has happened to the hoisting apparatus. So great is the confidence of the men who install these safeties that they have been known to run the car from top to bottom of a high building, regulating its speed entirely by the safety jaws, without a rope on the car.

This, however, is depending too much on the personal element, as the downward plunge of an elevator car, with nothing but free air to check it, is sufficiently trying to some men's nerves to make them forget for a few seconds what to do with that little screw which is holding the safety off. If they let go of it, its pitch is steep enough to spin it out of its nut, but the natural instinct at such times is to hang on to everything rather than let go of anything. The writer does not believe in making it possible to release the safety from the inside of the car, but there are others who insist upon it, and the insurance companies do not object.

We now come to that form of elevator safety that allows the car to fall unretarded till it reaches a certain distance from the bottom of the hoistway, where it is brought to a gradual and safe stop. But this, as before stated, is used only in addition to, and never as a substitute for, any of the safeties already described.

There is but one form of this device worthy of consideration; that is the air cushion formed by the lower part of the hoistway. The value of this device was first discovered by Mr. Albert Betteley, of the firm of Williams Adams & Co., of Boston, in 1857, when one of his elevators in the State Street Block in Boston fell to the bottom, loaded with boxes of sugar, and no injury occurred to the boxes. As the car had fallen from a considerable height with nothing apparent to check its speed, Mr. Betteley was led to look for some unusual cause for this successful fall, and discovered it in the comparatively air-tight construction of the bottom of the hoistway.

Until recent years, before buildings exceeded six or eight stories in height, these air cushions were usually made by constructing a strong pit below the lower landing of the car, about ten feet deep, with a variable clearance around the car platform at the top and as little clearance as possible at the bottom. This pit room below the lower floor is not absolutely necessary, as the following view, Fig. 280, of an Ellithorpe air cushion shows:

This device cannot be described as other than the Ellithorpe air cushion, for, although Mr. Ellithorpe did not fundamentally invent it, he was the first to drop voluntarily into one, over twenty years ago, and he has been doing it every year ever since, and he has so covered it with detail patents of his own invention that it is certainly his own device, morally, physically, mentally, and, we hope, to his profit financially.

It is highly probable that the Ellithorpe air cushion will be specified in the building laws of all cities for elevators carrying passengers.

As a modern installation of this device I cannot do better than to quote from the *Iron Age* of August 4, 1898, and reproduce the illustrations printed in that article:

"Even with all the experience and skill which have been devoted to the study of elevator safety appliances, with the best material and workmanship, with the most rigorous and continuous systems of inspection, and with competent persons in charge, yet passenger elevators sometimes fall and cause more or less serious accidents. The manufacturer of elevators uses the best and most efficient safety devices he can obtain to control the movement of the car and to surely arrest it if a certain speed should be exceeded. The very nature of his business compels him to do this, because the result is financial embarrassment to him if his elevators drop occasionally. This applies with equal force to owners of buildings, who would have difficulty in securing tenants if the elevator apparatus were suspected of being dangerous. Many even go beyond the purely mechanical device and introduce a pneumatic arrangement as a last resort—only to be brought into action when all else fails.

"The air cushion, located at the bottom of an elevator shaft, possesses peculiar inherent advantages which cannot be gainsaid. First and most essential, it is always ready instantly to

FIG. 280.

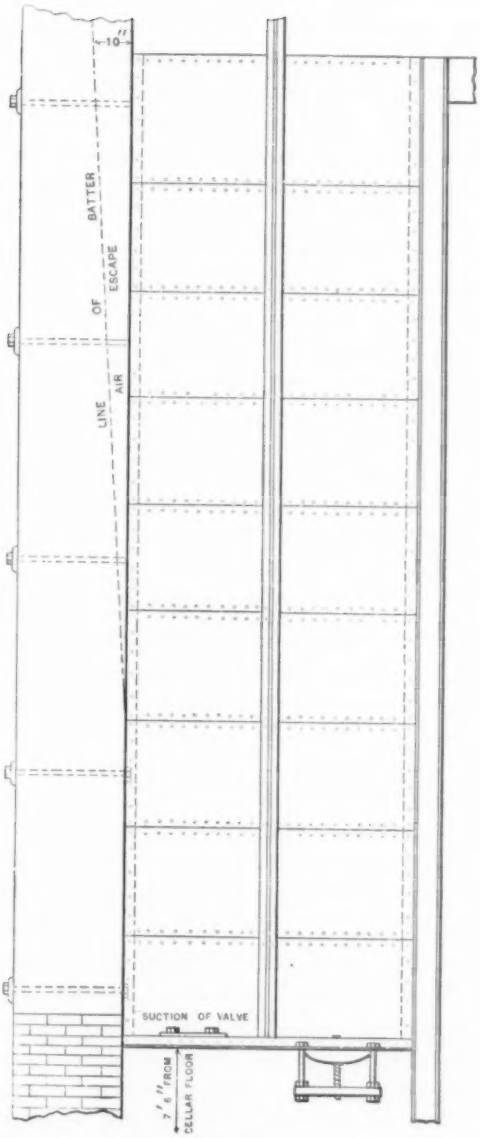

10"
BATTER OF ESCAPE AIR LINE
SUCTION OF VALVE
7' 6" FROM CELLAR FLOOR

FIG. 281.

perform its work and to do it successfully under all conditions. Of itself it cannot get out of order, since practically it is only a hole into which something may drop sometime. Whether the car dropped one or twenty stories its movement would cease, not suddenly, but gradually and without shock. The first cost of the air cushion is small and the outlay for its maintenance *nil*. It occupies space not otherwise valuable. All things considered, it is difficult to understand why it is not more widely employed.

"One of the most extensive and elaborate applications of the elevator air cushion is to be found in the Empire Building, at Broadway and Rector Street, New York, designed by Kimball & Thompson, Manhattan Life Building, New York. The elevators were installed by Otis Brothers & Co., New York, and the air cushions were designed by F. T. Ellithorpe, 136 Liberty Street, New York.

"The building is a twenty-story office building, recently completed and provided with all the most modern appliances and conveniences. There are ten elevators, of the high-speed hydraulic type, arranged in two groups of five each, one group being shown in sectional plan Fig. 282. While nine of the elevators are distinctly for passenger service, one is more powerful and is capable of lifting safes weighing 8,000 pounds. Each shaft is entirely independent from the floor of the third story to the bottom and is inclosed by walls which are not perforated except by the door openings. This forms the air cushion proper, which, as indicated in Fig. 281, is about 50 feet in depth. The doors of the main floor, Fig. 283, and of the second floor are in two parts which slide in recesses in the wall. They are of bronze and of ample strength to resist the air pressure that would come upon them if a car should fall. The usual open ironwork is entirely absent on these two floors, solid masonry replacing it. The cars have also been strengthened with the view of resisting this pressure. By consulting Fig. 281 it will be noticed that the shaft walls are battered for a short distance below the third-story floor. This provides a graduated air escape and adapts the cushion to any fall which the car may make. The car fits more closely in the lower portion of the shaft, the walls of which are vertical. It has been estimated that the air cushion should be in proportion of 1 to 6 of the travel; in the present instance the cushion is 50 feet and the

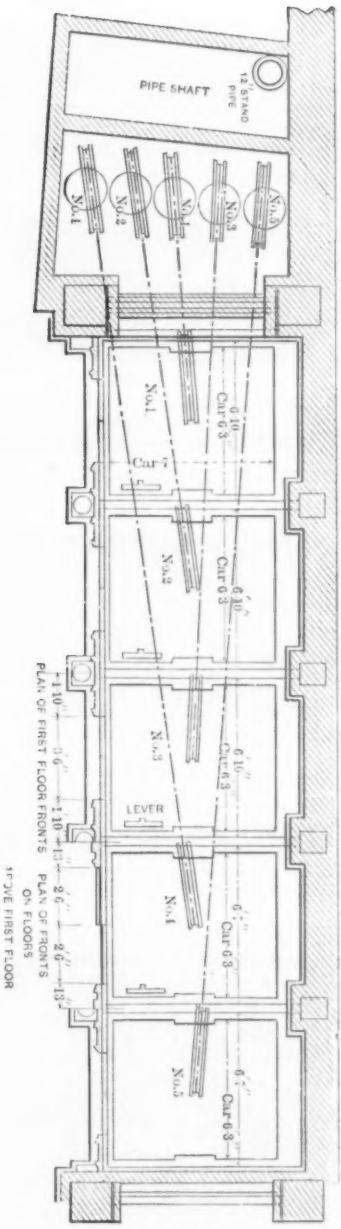

PIPE SHAFT
12" STAND PIPE
No.4
No.2
No.1
No.3
No.5
No.1
Car 6'3"
6'10"
Car 6'3"
No.2
Car 6'3"
6'10"
No.3
6'10"
Car 6'3"
LEVER
No.4
Car 6'3"
No.5
Car 6'3"
1'10"
1'10"
2'6"
2'6"
PLAN OF FIRST FLOOR FRONTS
PLAN OF FRONTS OF FLOORS ABOVE FIRST FLOOR

FIG. 282.

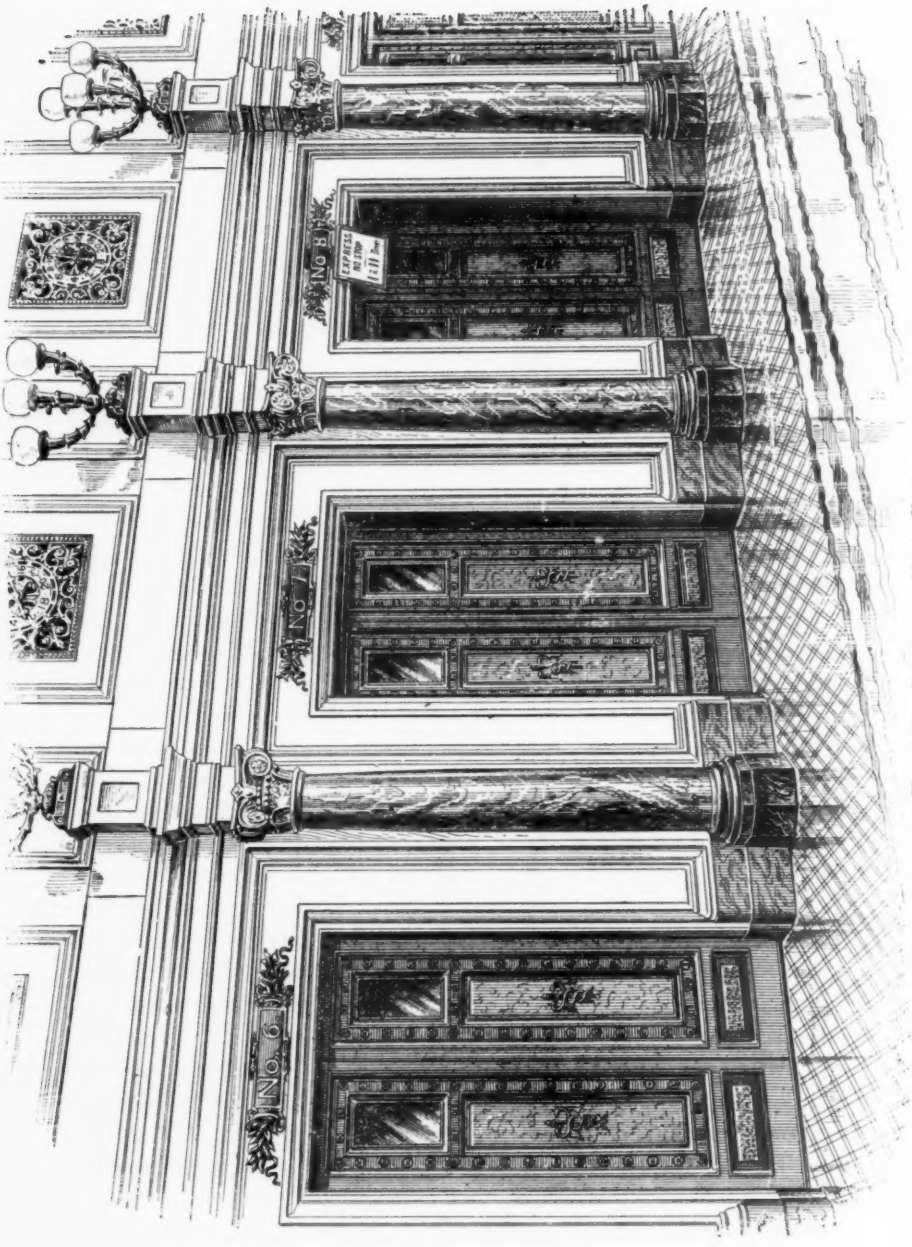


FIG. 283.—DOORS OF MAIN FLOOR

travel 287 feet. In the bottom of each shaft is a suction valve which opens inwardly as the car ascends, thus preventing the vacuum which would result from the car leaving the cushion. There is also an Ellithorpe improved escape valve, Fig. 284, which opens outwardly into the atmosphere. It is so adjusted as to sustain the weight of a car under ordinary conditions, but will, in case of accident, relieve the cushion of undue pressure when the car falls. It has been calculated that the pressure in the air cushion if a car should fall from the top would be 3½ pounds to the square inch.

"On July 18 a car weighing 2,000 pounds was dropped from the twentieth story. The efficiency of the cushion was shown

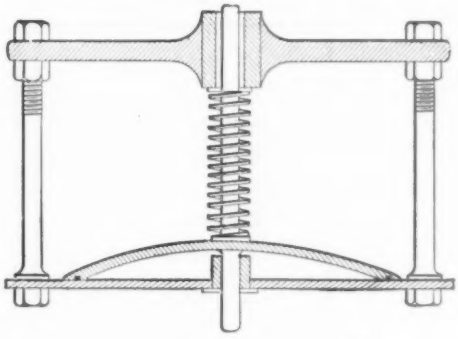


FIG. 284.

by the fact that the eggs and incandescent lamps carried upon the floor of the car were uninjured."

Any subsequent fall should be like the first, as there are no joints, packing, or automatic devices to be relied upon to assure the reliability of this device, the car has inches of clearance in its dash-pot action, the air pressure varies with the velocity of the car, and the velocity of the car varies but little with the load. The first or experimental drop in the air cushion will be exactly like every other drop the car may make into it, and nothing but unmitigated vandalism could render it inoperative.

From an elevator builder's point of view I have given too much attention in this paper to the subject of elevator-car safeties, and this is due to the purely commercial consideration that one safety is as good as another and no preference given to the

bidder having the best one. Accidents are in fact so rare in proportion to the number of elevators in operation that there is really cause for this indifference, which has been largely responsible for the general inefficiency of some of the cheaper forms of safeties.

A striking exhibition of this indifference is the cool way in which engineers of buildings tie up their car safeties after they have had to release them when they have been accidentally operated, and calmly let the cars continue to run indefinitely without them.

Within the last few years, however, a demand has been created for a reliable safety, and this, as with any other reasonable

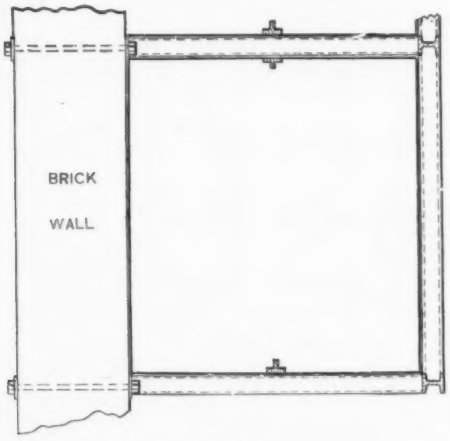


FIG. 285.

demand ever made upon a mechanical engineer, has been promptly supplied; if the demand had been made before, it would have been as promptly and as well supplied as it was later. High buildings, greater speed, and more elaborate hoisting apparatus called for a better safety, which was made as soon as it was wanted.

In the beginning of this paper I classified some of the salient features of elevators, but have only thus far dealt with the one subdivision of "Safety," which involves the car alone. As the balance deals entirely with the hoisting apparatus and is too much involved to treat in the same specific manner, I will endeavor to describe in order of their evolution the various types which have been in general use.

The generic trade classifications of these types are Drum, Hydraulic, and Screw. "Drum" implies all hoisting machines in which the ropes or chains leading from the car are fastened to and wound upon a drum. "Hydraulic" includes cylinders having their pistons connected to the car by ropes, or acting direct on the car, when they are called plunger elevators. "Screw" is the trade name for hoisting machines transmitting motion to the car by means of a nut and screw.

Drum elevators are the earliest, the most universal, and the most simple form of elevator ever built. The drums were first driven by worm gears, in the first instance for safety and smooth motion only, but the low efficiency of the early forms of worm gears caused some makers to use a spur-gear drive, which is more expensive if well built and never as safe or as smooth running as the worm gear. Modern improvements in worm gearing have raised its efficiency from 30 per cent. to over 80 per cent. in elevator gearing, and, although it can be built for less cost than spur gears and is equally as efficient, there are some concerns still making spur-gear elevators.

The drum elevator is especially convenient to overbalance, and can be placed in the top of the hoistway with one face of the drum plumb over the centre of the car, and the other face over the counterbalance, the car ropes fastened at one end of the drum grooves and the counterbalance ropes at the other end of the same grooves, the car ropes occupying the entire surface of the drum when the car is up, and the counterbalance ropes following alongside of them occupying the entire drum surface when the car is down. The counterbalance is made to balance the car with its average load, in which case the elevator has no gravity work to perform, and makes the most economical type of hoisting apparatus.

This arrangement, however, of placing the machine in the top of the hoistway is peculiarly adapted to electric elevators, when the power is easily conveyed by electric conductors to the motor, and was not always practicable with belt-driven elevators, and seldom used with steam hoisting engines. But whether placed in the top, bottom, or remote from the hoistway, the drum machine is most always overbalanced; even if it takes a dozen sheaves to do it, the saving in gravity work compensates for all friction and first cost so incurred.

There is but one radical objection to a drum elevator, and

that is the lack of absolute means of stopping its hoisting power when the car or counterbalance gets to the top of the hoistway. There is not one drum elevator—and by this I do not include the friction-drive cable-drum type, which is not in general use, unfortunately—that is not liable to run its car or counterbalance into the top of the hoistway and cause an accident. Automatic devices are operated by both car and hoisting machine to shut off the power and apply the brake at the limits of the car's travel, but they are all adjustable, removable, and breakable, and nothing but the eternal vigilance of the personal element can assure their proper action. This is not the case with the hydraulic or screw elevators, as the car travel in both of these types is limited by the travel of the piston or the nut, independent of shutting off the power. Neither is it the case with a friction cable drum drive, which is simply a vertical cable railroad with an elevator car on one end of the ropes and its counterbalance on the other, for dumb-waiters only a single bight of the rope over the driving sheave giving sufficient friction; and for general elevator work, two driving drums geared together, or one driving and one idler drum with shafts at proper angles to each other to get as many driving turns as may be required. Gravity tension at each end of the ropes supplies the friction of the cable-road tension sheave, and buffers being provided for the car and the counterbalance, to be stopped at the bottom before either one reaches the top of the hoistway, the hoisting effort is positively arrested by the slacking of the ropes on the driving drums the moment either car or counterbalance arrive at the bottom.

This same function also prevents the paying out of slack cable when the car is stopped in its descent by other cause than its hoisting machine. In some types of drum elevators, delicate or crude and unreliable devices are used to stop the hoisting machine by the action of the ropes when slacked by the car being so stopped. The writer has used the action of the car safety on drum elevators electrically controlled, to stop this paying out of slack cable, the safety being the usual cause of slacking the car ropes, and is the most reliable means of operating a slack cable device. Hydraulic elevators, and the screw elevator invented by the writer, depend upon the tension of the hoisting ropes to maintain their motion during the descent of the car, and thus normally avoid this slack-cable trouble.

The form of worm gearing to be used on drum elevators involves the following conflicting considerations:

Efficient angle of worm increases cost and diameter of worm wheel or the cost and size of electric motor, difficulties of electric control, and the danger of running down at too high speed. Hindley worm gears are more durable and efficient than any other style, but they require more elaborate tools to make them and more accurate alignment to run them, and their durability and efficiency in actual practice do not seem to exceed that of other styles enough to compensate for the extra trouble that they involve. Their direct cost of manufacture does not exceed that of other forms of worm gears, and their alignment can be successfully accomplished with proper tools, and maintained by proper bearings. As they can transmit more power than any other worm gears of equal size, their use could properly be confined to hoisting machines requiring great capacity in a small space.

The interlocking right- and left-hand worm gears, taking up their own thrust, were better than a single worm and gear until ball or roller thrust bearings were perfected as they are to-day. Now, with a roller thrust having less friction loss than a worm and gear, the efficiency of this double-worm gearing cannot be as great as a single-worm gear with roller thrust, provided the single-worm gear is not overloaded.

The writer was the first to design and use right- and left-hand Hindley worm gears, interlocked by spur gears bolted to them, as the Hindley gear teeth are straight like the thread of the worm, their only curve being to conform to the helical surface of the worm, and therefore cannot mesh together. All other forms of double-worm gearing mesh the worm gears together; their angularity being reversed as the worms coincide where they mesh together like spiral gears, some using unfinished cast teeth and others regular involute teeth cut straight across at the proper angle, with faces wide enough to admit a small-diameter hob cutting a surface for the worm in the centre of the faces and leaving enough for spur-gear drive at the ends.

To make a perfect job, these teeth should first be cut by a spiral gear cutter, but I do not know of any so made, and, like the Hindley gear, it might be an ultra-refinement for commercial use. However, as in all other problems in mechanics, we know what is best long before we obtain it.

The double Hindley geared elevators referred to are clearly shown in the following drawings, Figs. 286 and 287.

They certainly appeal to mechanical engineers, and such plants as the Siegel-Cooper store in New York City and the Central London Railway in London have selected this type on account of its superior durability; the last-named plant using 49 of these elevators, operating cars having floor areas of from 117 to 256 square feet each, averaging 13 tons total load on the ropes for each car.

As before stated, there are some objections to an efficient angle of worm, some of them being due to commercial limitations of electric motors. We may as well recognize the fact that for the future no other form of power need be considered for this type of elevator; probably 90 per cent. of them are now being built electric.

The conditions limiting the angle of the worm are the diameter of the gear, the speed of the motor, and the ability of controlling the speed of a car that can easily drive the worm gear.

The diameter of the gear is limited by the diameter of the drum, and the diameter of the drum is limited by the range of work intended for the particular machine, which in general practice does not exceed 36 inches for the smallest drums used for the heaviest loads that the machine has power to handle; the lighter loads requiring higher speeds are operated by drums of larger diameter. Another important limitation to the diameter of the gear is the cost of the gear itself and of all parts of the machine that would be increased in proportion. Such a gear would in turn limit the variation of drum diameters, as drums smaller than the gear require extra idler sheaves to lead the ropes up between car and hoistway when the machine abuts against the hoistway wall.

The gear for the average drum elevator is therefore practically limited to a diameter that will, with its gear case, not extend beyond the flanges of a 36-inch drum; such gears are about 30 inches in diameter. Having determined this, we can now consider what angle of worm can be obtained at the required car speeds and limitation of motor speeds. The duty on drum elevators ranges from 2,000 pounds at 300 feet per minute to 6,000 pounds at 100 feet per minute, varied by drums from 36 to 60 inches diameter, and motor speeds from 470 revolutions per minute to 1,000 revolutions per minute. This

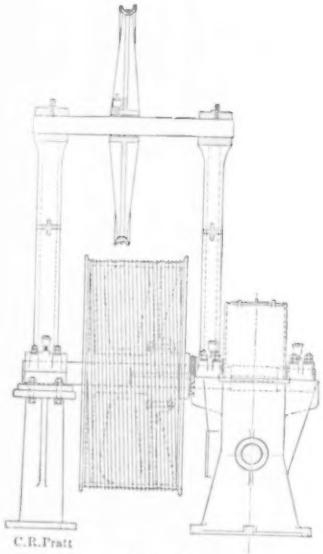


FIG. 286.

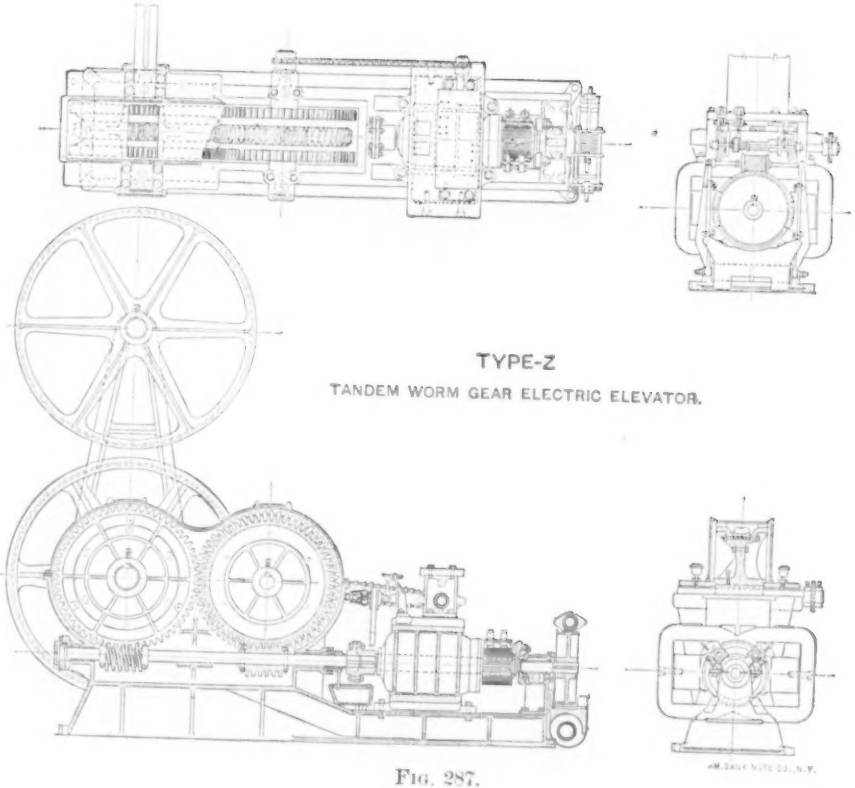


FIG. 287.

of course, covers differences in worm-gear reduction arbitrarily determined by the designer.

Assuming 470 revolutions per minute as the lowest motor speed consistent with the cost of a motor of the required horse-power, the cost being in inverse ratio to its speed, we get a reduction of 46 to 1 to obtain 100 feet per minute on a 36-inch drum, 24 to 1 for 300 feet per minute on a 60-inch drum; and a reduction which the writer has found to be very successful is 29.5 to 1 for 200 feet per minute on a 48-inch drum. This latter is that shown in the Hindley gears in Figs. 281 and 282, and has an angle of 12 degrees, using a double-threaded worm and 59 teeth in the gear. Seventy per cent. efficiency has been obtained with this gear from current delivered to motor to work done at car, giving the gear over 80 per cent. efficiency. One elevator with this gear is being run as what we call a gravity machine; *i.e.*, the car is under-balanced and drives the motor connected up as a dynamo in its descent, starting promptly as soon as the brake is released. While it has been considered dangerous by most elevator builders to use a gear that the car can drive, the writer has no reason to doubt the safety of this hoisting machine, which has a brake of ample capacity, applied by a spring and released by a magnet, which lets go and applies the brake whenever the electric current is broken, either by an overload or by a switch operated by a centrifugal governor when the speed exceeds that which it is regulated for.

While this particular instance obtains an efficient angle of worm, the same conditions do not always exist in drum elevators. In the first place this high efficiency is in part due to the Hindley gear; and, secondly, the low motor speed is commercially due to the horse-power of the motor, which is the greatest that is used on drum elevators, speeds of 800 revolutions per minute being more often used than 470 revolutions per minute; for these car speeds and gear diameter reduce this angle considerably, but not in the same ratio as the revolutions per minute reduction, because with the reduction of load on the worm its diameter can be reduced and a greater angle maintained, usually between 8 and 9 degrees, and an efficiency of about 70 per cent.

Recent data on worm gearing indicate 45 degrees as the angle of maximum efficiency, and a curve plotted from the data obtained rises abruptly to about 15 degrees and then flattens

out to 45 degrees, which gives these elevator gears a low rating.

The angles here given of 8 and 9 degrees are those generally used in the best elevators on the market; there are many others in use where the angles are less than 6 degrees, and the total output of such elevators is not over 30 per cent. of the current used.

Chronologically the hydraulic elevator follows the drum type, but antedates by many years the electric drum elevator.

The simplest form of hydraulic elevator is that called the plunger type, which consists of a ram working in a vertical cylinder with stuffing box at the upper end. While this has been exploited for high rises in a telescopic form, its practical use is for sidewalk elevators, or elevators where the car is only a platform rising to a floor which permits of no continuation of the guide rails or car frame, in which case the stability of the ram holds the car platform level under unbalanced loading. Such elevators seldom rising over 30 feet, no telescopic arrangement is necessary, and the cylinder can be bored its full length into the ground to allow the platform to descend to the lower floor.

There is but one other class of hydraulic elevator, aside from those which were unsuccessful experiments, or which are now being experimented with, and this is built with either a vertical or horizontal cylinder as is best suited to the building; the vertical cylinder, being the better type, is always used where it can be installed. The following views, Figs. 288 and 289, illustrate the construction of this elevator very clearly.

The motion of the piston is controlled by the exhaust port, the pressure being always on top of the piston; raising the piston valve opens the exhaust and leaves pressure on the top of the piston only; lowering the valve admits pressure on both sides of piston, and the load raising it circulates the water from top to bottom of cylinder through pressure and exhaust ports. The piston valve, as here shown, closes the exhaust port from both pressure and discharge connections, holding the piston between a closed port on one side and the pressure on the other. The main piston partly cuts off the ports at each end of the cylinder, to check its speed before striking the cylinder heads in case the piston valve should fail to work. This piston valve is controlled from the car either by a shipper rope direct; a shipper rope connected to a pilot valve, which in turn operates

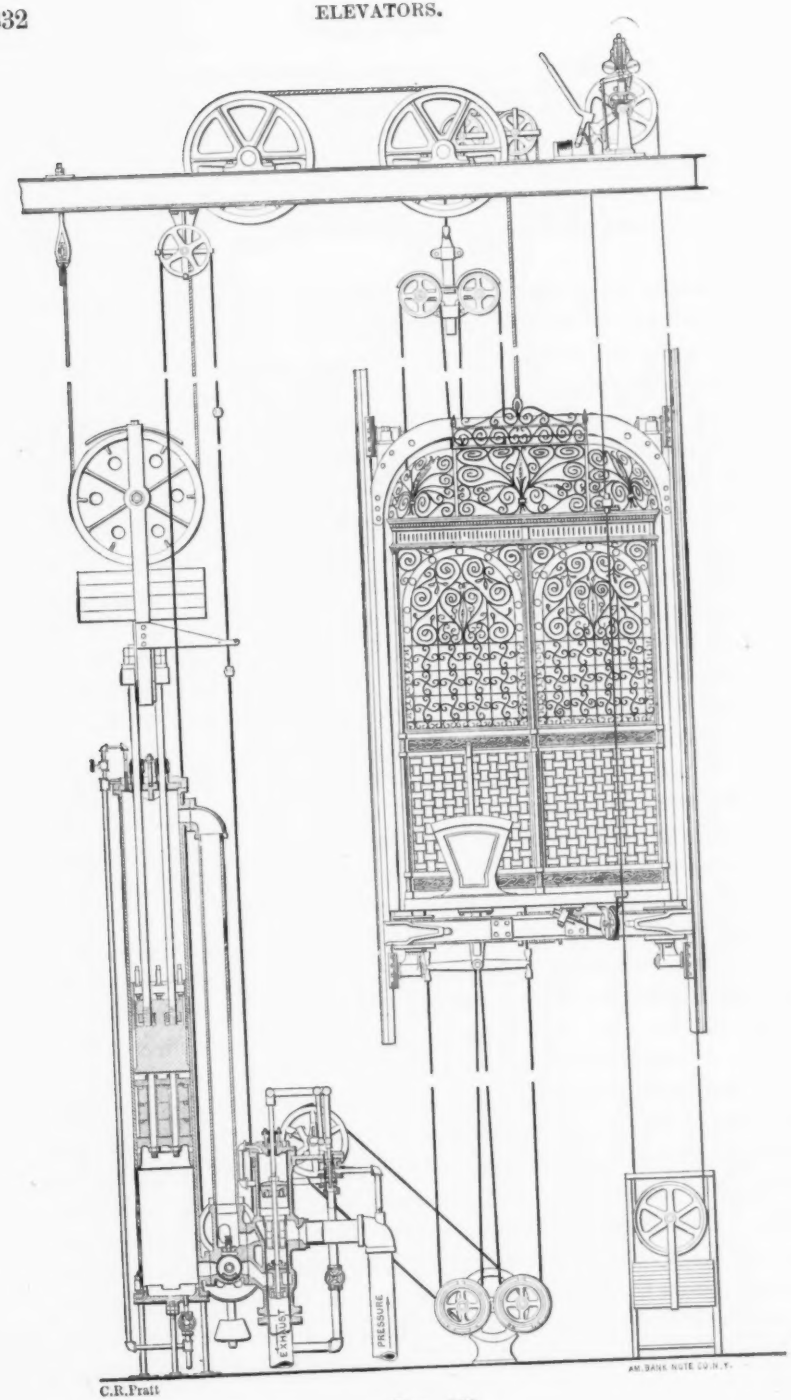
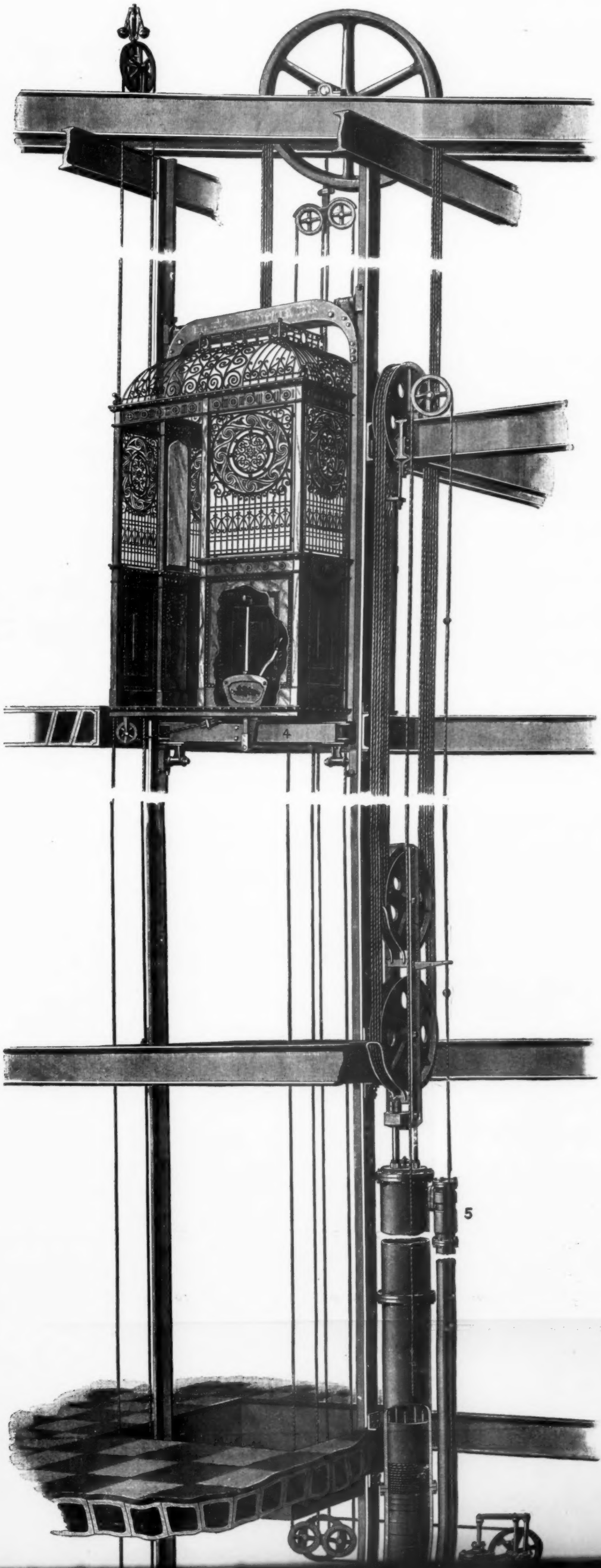


FIG. 288.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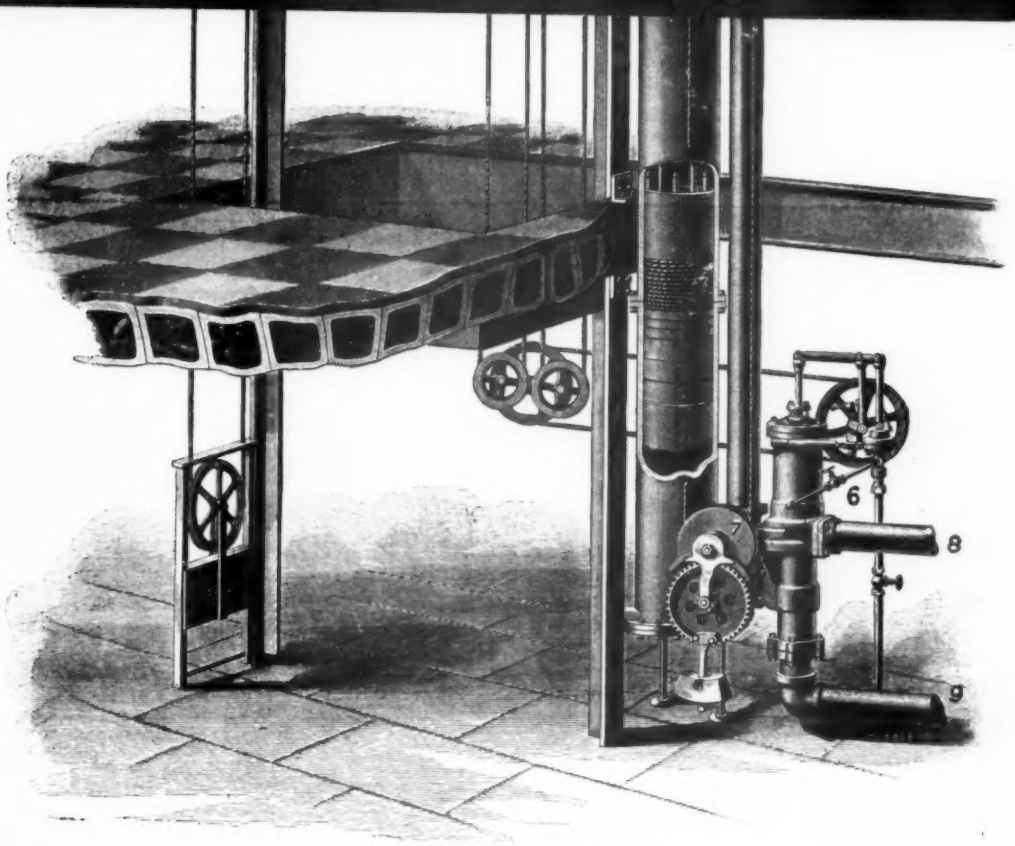


Fig. 289.

the piston valve; or a travelling shipper rope, operated by a lever in the car, as shown in Fig. 288, is connected to a pilot valve operating the main piston valve. To stop the car at the limits of its travel a shipper device attached to some part, moving with the main piston, closes the piston valve at the proper point of the car travel.

The object of maintaining a circulation of water from top to bottom of piston through the pressure pipe and valves is to balance the atmospheric head of water and leave the effective pressure on the piston the same during all parts of its travel, which can be done up to 30 feet piston travel.

This is not needed in horizontal cylinders, and they therefore have water on one side of piston only, using the same port for pressure and discharge, and a drip pipe on the other side of the piston.

There are two general types of these horizontal cylinders, one having tension and the other compression piston rods. The tension rods, being on the water side, require stuffing boxes, while the compression rods, usually in the form of one or two large wrought-iron pipes, are on the open end of the cylinder and require no packing.

These cylinders are usually of large diameter, shorter and higher geared than the vertical type. By higher gear I mean a larger number of multiplying sheaves, which in horizontal machines usually multiply the piston travel by 10, giving 10 feet car travel for every foot of piston travel.

The advantage of the vertical cylinder over the horizontal is in the counterbalancing effect of the piston and parts moving with it upon the car, which, being carried on a solid column of water, do not tend to teeter the car as a free counterbalance does when the car is stopped in its ascent and the momentum of the free counterbalance raises the car after the hoisting power has been cut off. Such counterbalances easily raise a car, having a speed of 500 feet per minute, 6 feet beyond the point when the hoisting power cuts off, and of course let the car fall back that distance and teeter (to use an elevator term) several feet up and down before coming to rest. This is only avoided by fine valve adjustment and taking plenty of distance to stop in.

Another cause of teeter is the whip of the ropes on horizontal machines, which is not common to the vertical; this is greater

in the compression than in the tension piston rods, as the ropes on the former pass around the cylinder from the multiplying sheaves anchored at the water end to the travelling sheaves at the other end, or nearly double the distance of the tension piston-rod type, where the travelling sheaves meet the anchored sheaves at one end of their travel.

We must pass over valve design and construction as matter involving too much detail for the scope of this paper: enough to say that no branch of hydraulic machinery has developed the variety of ingenious devices employed in controlling hydraulic elevators, and by no other means can an elevator car be better handled.

To avoid transmitting pump pulsations to the car, and also for storing power to use after the pump is stopped, gravity tanks, pressure tanks, gravity-pressure tanks, and accumulators are used.

The gravity tanks which used to decorate the roofs of buildings are seldom used now; the pressure tank in the engine room, as shown in Fig. 290, came next, accumulating air pressure over the water line for any amount required, air being mixed with the supply by a valve to maintain the proper water level. Placing this tank on the roof as a gravity-pressure tank permitted the air from the cylinder to escape more readily and gave greater storage of power by using gravity pressure for light loads as the air pressure reduced.

Accumulators are chiefly used with high-pressure plants, but pressures over 250 pounds per square inch are not often used.

For pumping water into these tanks all kinds of pumps are used, electric or gas engine power pumps, single-cylinder, duplex, compound-duplex, and high-duty pumping engines, according to the size and character of the plant; compound and high-duty pumps being economical only where enough elevators are running to give the pumps a steady load.

The following reproduction, Fig. 291, of a photograph taken at the Whittier Machine Company's shop in Boston, shows an electric elevator invented and built by the writer in 1888 and 1889, and, so far as the writer or those associated with him at the time knew, the first electric elevator ever built.

I had simply taken hydraulic-elevator sheaves, mounted them in a suitable frame, and pulled them with a nut and screw instead of by their usual hydraulic cylinder and piston. The

screw was driven through a double spur-gear reduction by the Thomson-Houston electric motor seen in the right of the photograph. The screw had two threads of one-half inch square

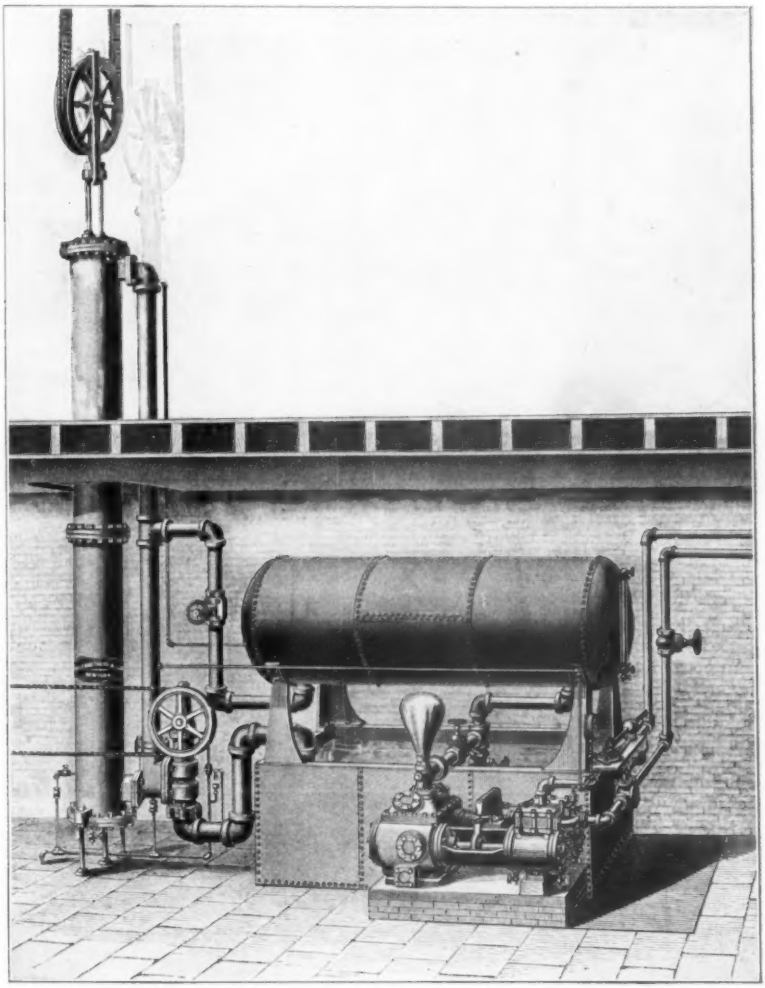


FIG. 290.

section, 2-inch pitch, and 4-inch outside diameter; it carried a bronze nut covering twelve threads. I was told by both Thomson-Houston and Sprague electric engineers in Boston that it would never be safe to start an elevator car coupled direct to an electric motor, so I built a centrifugal clutch and brake

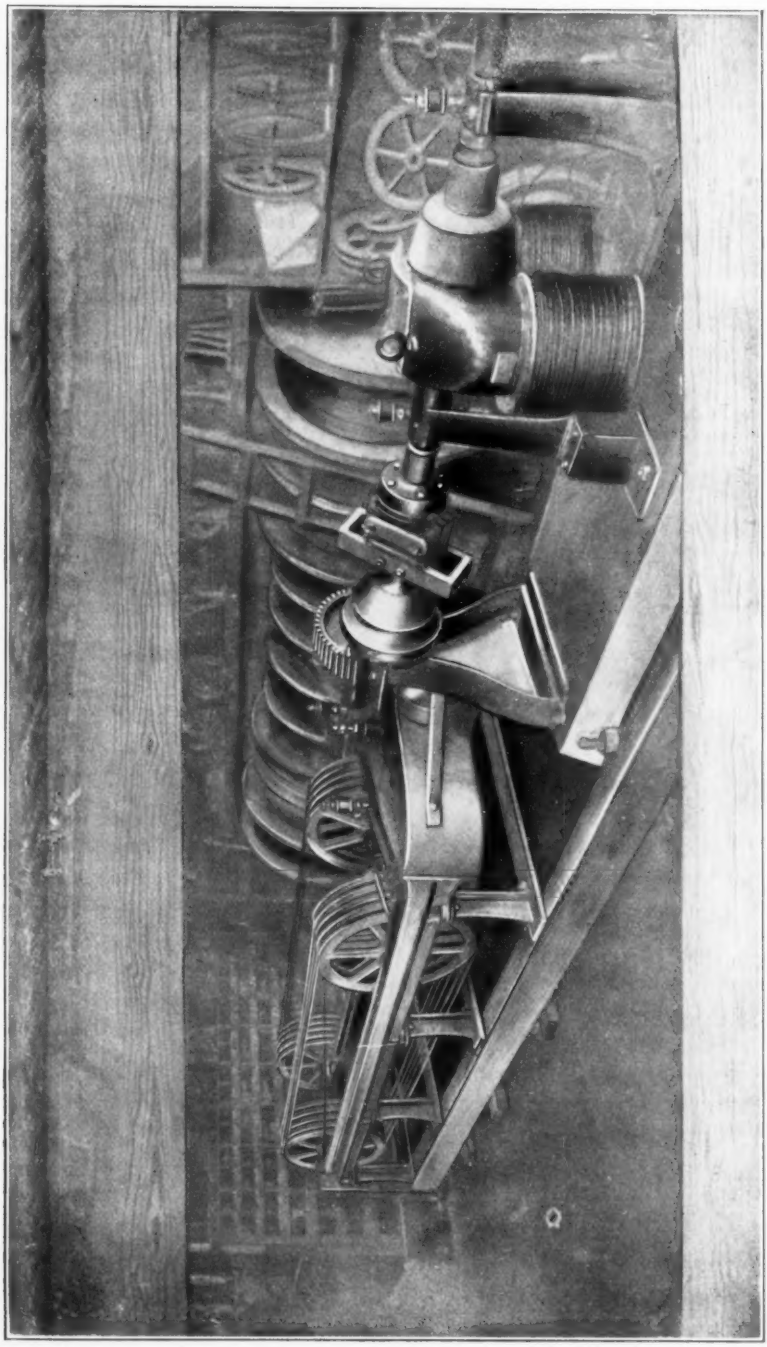


FIG. 291.

permitting the armature to revolve freely till it attained a speed of 200 revolutions per minute, and then engage the load and release the brake at the same moment by friction driving clutch and brakeshoe; this it did, the first and every other time, making a very easy start and stop, but was about as necessary an attachment to an electric motor as a safety valve would have been. The motor was controlled by a street-car rheostat on the car, carrying the line current.

My next discovery was a ball-bearing nut built by Mr. C. A. Lieb, and exhibited by him as a curiosity of no practical value. I did not agree with him as to its practical value, and arranged to use it on my next elevator, redesigned to overcome the difficulties experienced by Mr. Lieb.

This elevator, called the Sprague-Pratt electric elevator, was not put in actual use until September, 1891, when it was located on the fourth floor at No. 135 West 23d Street, New York City, as shown in Fig. 292, and connected to the only passenger car in the building by a separate set of ropes, to enable us to change back to the old steam hoistway machine to run the car by that when we wished to lay ours off for alterations or repairs.

On May 19, 1892, at 6 A.M., a fire in the floor underneath this elevator burned the flooring from under it, dropped the plaster from above it, turned the fire hose upon it, and left it resting on charred timbers, covered with mortar and rust. At 8 P.M. on the same day it was in operation again, carrying passengers the same as if nothing had happened to it.

It was not absolutely fireproof; the next fire in this building finished its career, but by that time we had its successor in operation in the Grand Hotel, Broadway, New York.

The following views—Figs. 293 and 294—show the general design of this elevator. In Fig. 293 the ball-bearing nut in section is shown seated in the crosshead and having its centre supported by the crosshead guide shoe, the motion of the screw being transmitted through the balls to the nut, and from the nut to the crosshead, which carries the travelling sheaves.

Fig. 294 shows the single-deck horizontal type, and Fig. 295 the double-deck horizontal type; a triple-deck type is also built. The multiple-deck types are merely building independent machines one on the top of the other to save floor space. Referring to Fig. 294, the car ropes are seen leading down to the sheaves, which are carried on trunnions at left-hand end of the machine,

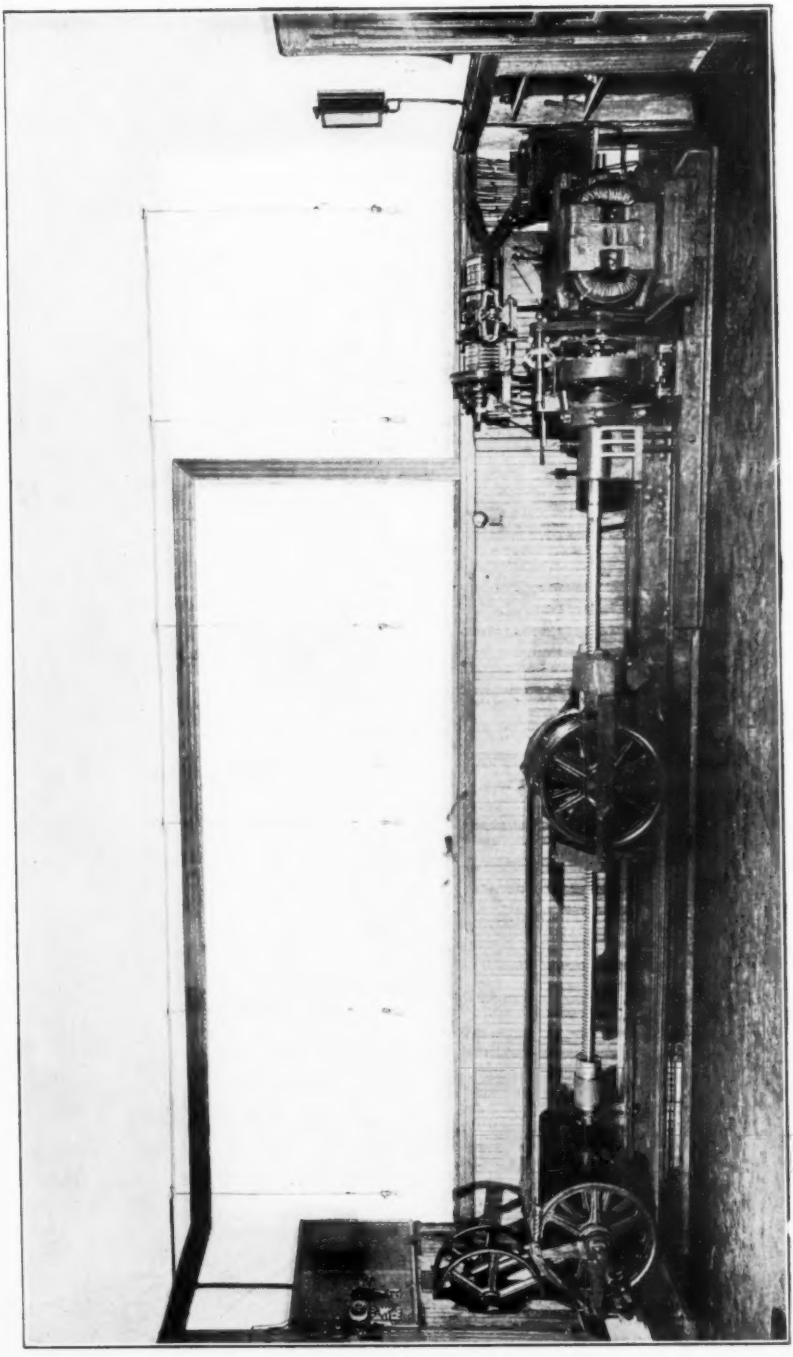


FIG. 292.

from there around the travelling sheaves, etc., until the desired multiplication is obtained, when they are anchored to an evener bar over the fixed sheaves. The usual multiplication is 8 to 1 on the machine and 2 to 1 on the counterbalance for high rise and high-speed cars. Four, five, or six $\frac{5}{8}$-inch ropes lead from the car over the overhead sheaves, down around a sheave on the counterbalance, up to and anchored at the top of the building four $\frac{3}{4}$-inch ropes lead from the bottom of the counterbalance to the machine, two of them passing around eight multiplying

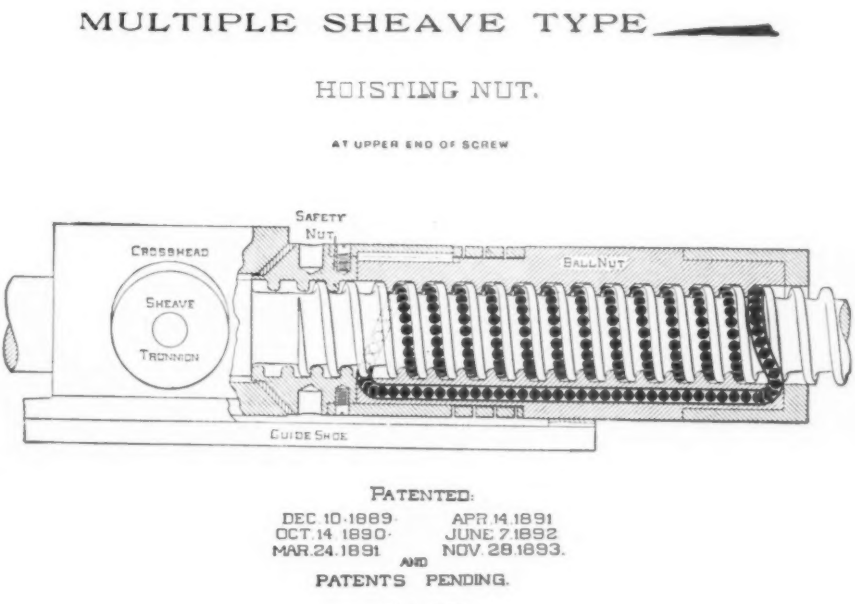


Fig. 293.

sheaves on one side of the machine, and two on the other side, making the travel of the car sixteen times that of the nut, and the load on the nut sixteen times the unbalanced weight of the car plus its load, which sometimes amounts to 80,000 pounds.

Figs. 296 and 297 show photographs of machines made in accordance with the drawings Figs. 294 and 295.

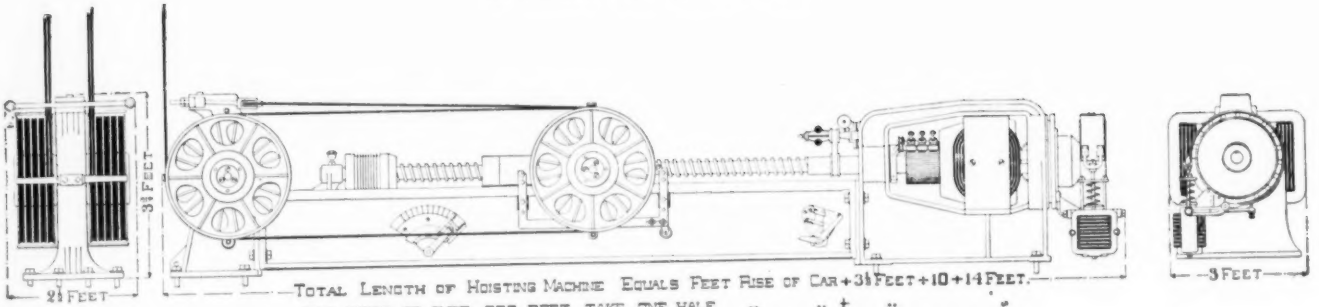
Referring to Fig. 293, it will be noticed that there are two nuts connected together by a splined sleeve, allowing them to be forced apart by a coiled spring shown between the sleeve and ball nut. Between the conical end of the safety nut and the crosshead is a conical bronze bushing, and the end of the safety

SINGLE DECK

TYPE E - 10 TO 1 - 4 R.

10 TO 1 = TRAVEL RATIO OF CAR TO NUT = 10 SHEAVES.

4 R SIGNIFIES FOUR HOISTING ROPES.



TOTAL LENGTH OF HOISTING MACHINE EQUALS FEET RISE OF CAR + 3½ FEET + 10 + 14 FEET.
FOR RISES OF OVER 200 FEET TAKE ONE HALF .. .. + ...
SPEED OF CAR WITH LBS LIVE LOAD = FEET PER MINUTE.

FIG. 294.

DOUBLE DECK

TYPE F - 10 TO 1 - 4 R.

10 TO 1 = TRAVEL RATIO OF CAR TO NUT = 10 SHEAVES.

4 R SIGNIFIES FOUR HOISTING ROPES

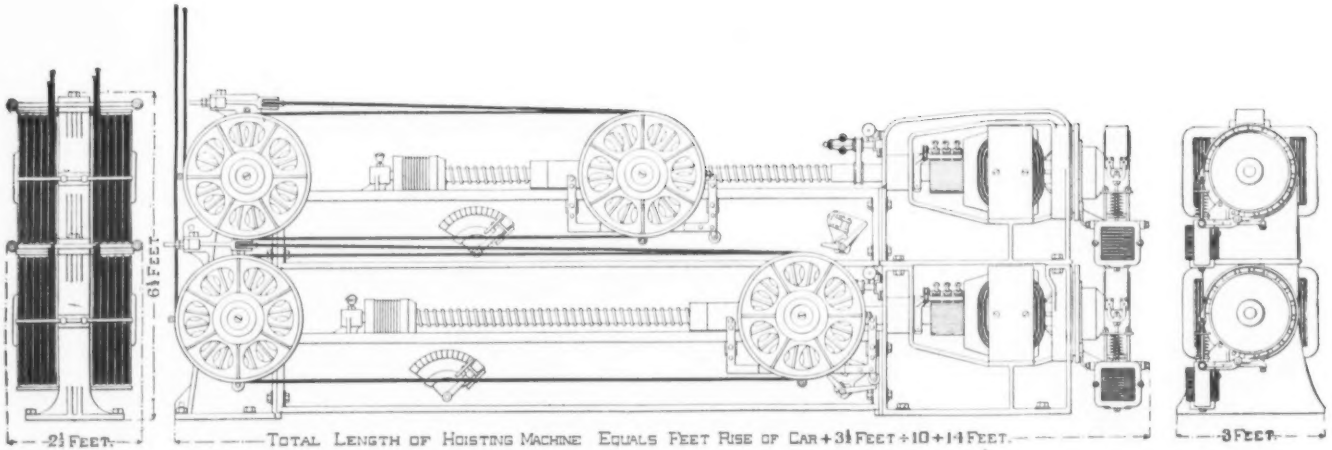


FIG. 295.

nut is shown in contact with a shoulder on the screw. This forms the upper limit of the nut's travel, at which point the car is about three feet above the top floor, and should, if the hoist way is properly designed for a high-speed car, be three feet below the overhead work. If the motor continues to revolve the screw after the safety nut meets this shoulder, on account of both operator and upper-limit switch failing to stop it, the nut simply revolves with the screw and turns upon its conical bronze seat in the crosshead; the friction at this point being always greater than between the nut, balls, and screw, holds the

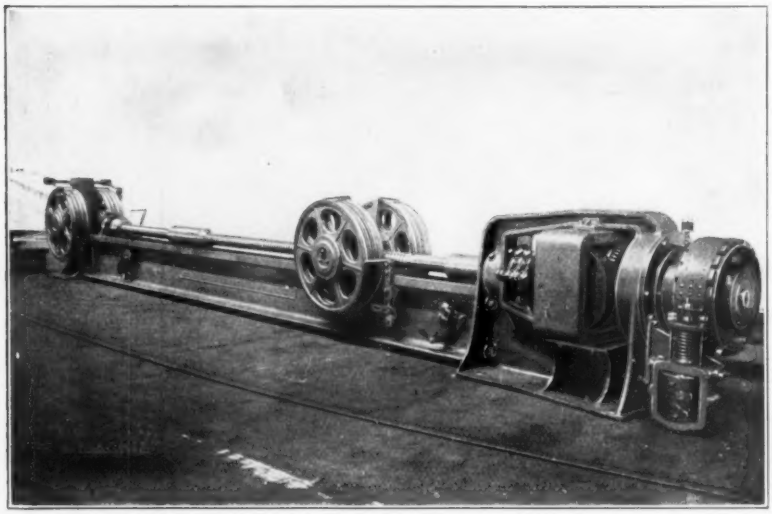


Fig. 296.

nut from turning while hoisting and lowering. At the left-hand end of the screw, as shown in Figs. 294 and 295, is a rubber buffer to stop the travel of the nut gradually should the lower-limit switch fail to stop it. These limit switches shown on the guide beam are operated by rolls on the crosshead frame engaging a bell-crank lever on the upper-limit switch, as shown on the lower deck of Fig. 295, and a cam keyed to the shaft of the lower limit switch and located on the opposite side of the guide beam from the switch.

Following the screw along to the right, we next observe a centrifugal governor belted to it; this operates a switch to open the brake circuit and apply the brake if the electric speed con-

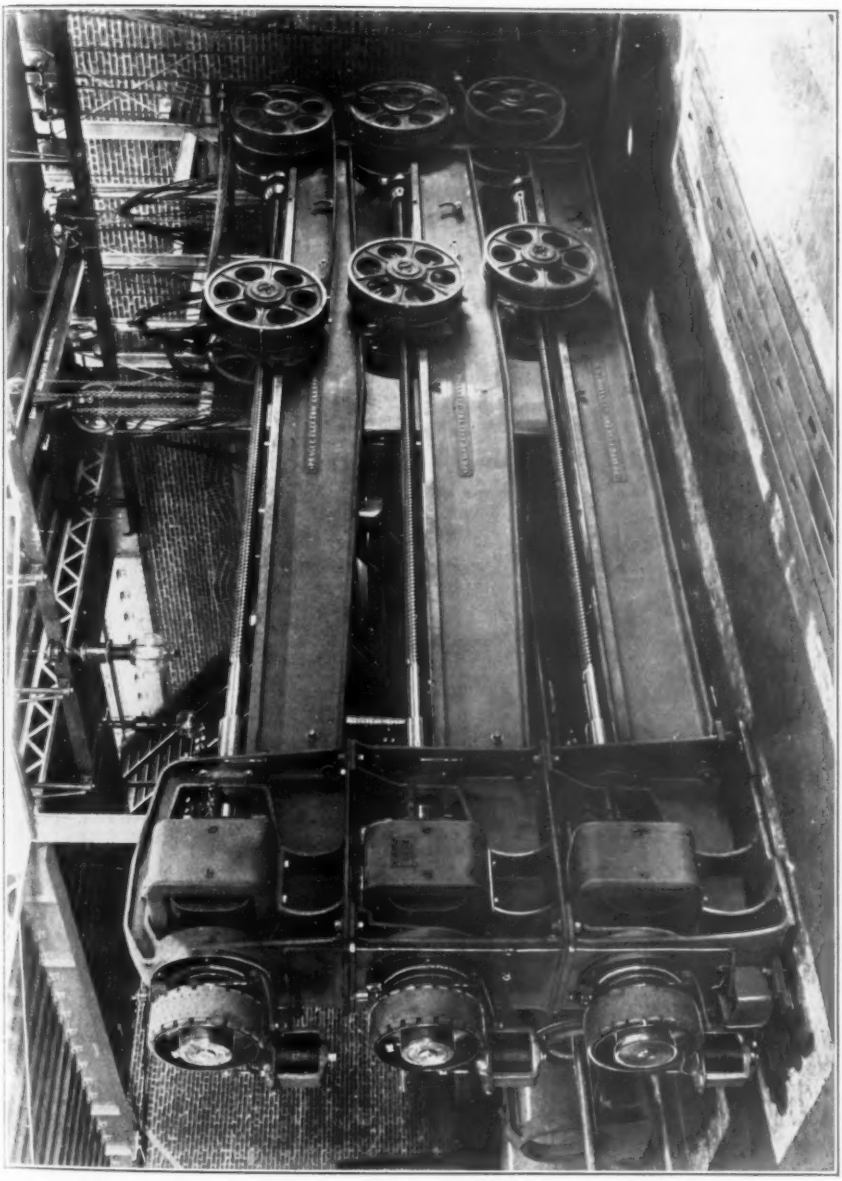


Fig. 297.

troller fails. As now built, this belt runs on the keyed taper coupling joining the screw to the armature shaft. At the time this cut was made the screw bar extended in one solid shaft through the armature to the brake pulley at the extreme right of the machine, but this made them too long to handle and too costly to make and renew.

The thrust of the screw is located on the right-hand face of the motor bed, placing the screw load under tension. The design of this thrust bearing is shown in Fig. 298, a photograph of the caged rolls which bear between two hardened steel plates 1 inch thick by 11 inches in diameter. It is interesting to note here that having gradually raised the gear of these elevators as our contracts called for higher rises and speeds, without enlarging the area of the nut or thrust to provide for double the load they were intended to carry, we had trouble getting a thrust within the dimensions that existing machines limited us to, that would stand these occasional loads of 80,000 pounds at 300 revolutions per minute, and average loads of between 20,000 and 30,000 pounds. We tried every arrangement of ball thrust with balls from $\frac{3}{8}$-inch to 1-inch diameter that was ever heard of, also every form of cone-roller thrust. Cone-roller thrusts would usually melt and weld in a few hours at the thrust of the cones, even when each cone had a ball-bearing thrust, and ball thrusts would wear out plates and balls in a few weeks. This state of things was getting pretty expensive, when in testing some soft balls in a thrust bearing made for that purpose I observed them form a straight rim and revolve on it with no tendency to form a crown. I took the hint and made a cage of 180 rolls $\frac{1}{2}$-inch diameter by $\frac{3}{16}$-inch face by $\frac{1}{4}$-inch thick, corners rounded $\frac{1}{32}$-inch, as shown in Fig. 298. This was placed in our greatest thrust-destroying elevator nearly two years ago; it has shown no measurable wear as yet and has acquired a most remarkable polish. Other thrusts of this design have failed on account of the plates not being perfectly hard, or defects in the rolls, but these are the exceptions.

The motor is a four-pole, rectangular steel-casting field, series wound, with just enough shunt to excite its dynamo action when driven by the descending car.

The brake, as shown at the right of the side elevation and adjacent end view, is a steel band lined with wood blocks, anchored at the bottom and gripping three-fourths of the pulley.

The tension is applied by a helical spring and released by a magnet, $\frac{1}{16}$-inch movement sufficing to clear the 20-inch diameter pulley.

Looking at the pulley end, the screw revolves from left to right in lowering the car, thus getting the cumulative friction tension of the band to hold the load.

The multiplying sheaves running on roller bearings places the entire transmission of power in this elevator on rolling friction, and attains the remarkable efficiency, for a machine of this character, of 70 per cent. from current delivered to motor to work done at car, giving the mechanical efficiency over 80 per cent. This is in hoisting. In lowering, the car load on the ball

FIG. 298.

nut revolves the screw, starting it instantly the brake is released, and drives the motor connected up as a dynamo, generating 50 per cent. of the energy of the descending load in current at the dynamo's terminals. This current, being a potential varying from zero to maximum, has not been utilized as yet.

Having a rheostat in which the highest resistance will just move the car without overheating while hoisting, and hold the motor on its dynamo circuit in lowering to a scarcely perceptible speed by very low resistance, all ranges of car speed are obtained.

As stated in the early part of this paper, this elevator, as well as the hydraulic, depends upon the tension of the hoisting ropes to maintain its motion in lowering, and requires no automatic slack-cable device to stop the paying out of slack cable when the car is stopped in its descent by anything but the hoisting machine. In the hydraulic, the rope tension on the piston is

the only power by which it can circulate water in the vertical or discharge it in the horizontal cylinder. In this elevator, the rope tension on the nut is the only power by which it drives the screw in lowering, but the momentum of the armature continues to revolve the screw after the nut pressure is released, and if the nut was fastened to the crosshead it would be carried to the end of its travel and slack all the rope on the machine; even if not fastened to the crosshead, its inertia would carry it a long distance on the screw before it turned with it. To compensate for the momentum of the armature and inertia of the nut, the nut is not only free to turn with the screw when the crosshead pressure is released, but the spring shown in Fig. 293 jams the safety nut back against the screw's thread and, forcing the nut to turn with the screw, prevents any slack cable.

Another function of this safety nut is to hold the load in case the ball nut's threads should wear out and strip, or rather to prevent this by being so placed in relation to the screw's thread that, while normally making no contact with the screw, it approaches the screw's thread as the ball nut's thread wears, and before it has worn to the point of stripping, the load is carried on the safety nut, which having too much friction to drive the screw and lower the car, the elevator is simply put out of operation till a new nut is furnished.

A vertical type of this elevator has been installed in the Park Row Building, New York City, where there are five elevators rising twenty-five stories, 297 feet, and five rising twenty-six stories, 309 feet, operated by these machines.

While the functions of this type are identically the same as the horizontal, the design of it, with the exception of the nut, screw, thrust, brake, and armature, is entirely different. The gear of these machines is sixteen to one, the same as used on horizontal machines for similar service. Suspended just below the lower limit of counterbalance travel are three pairs of machine multiplying sheaves, each pair being carried on a trunnion, and the three trunnions carried by two 1-inch by 10-inch steel bars, through which the ends of the trunnions pass with nuts and cotter pins on the outside. These 1-inch by 10-inch bars are suspended from beams resting on columns which extend down about 80 feet to the machine beds, and carry the 16-to-1 geared pull of the ropes.

Below the three fixed sheaves come four pairs of travelling

sheaves mounted in the same way, only their 1-inch by 10-inch bars are fastened at the bottom to two 3-inch rods which extend down to the nut's crosshead on either side of the screw. At the upper and lower ends of these bars are crossheads with guide shoes running on the same Tee rails that the counterbalance runs on; these rails extend from the top of the machine guides to the top of the building, and also keep the three fixed sheaves in line with the travelling sheaves. On either side of the counterbalance two $\frac{3}{4}$-inch ropes lead down close to the Tee rails to the lower pair of travelling sheaves, and pass upward from them to the upper pair of fixed sheaves, which are 3 inches smaller in diameter; from there down to the next set of travelling sheaves, etc., there being 3 inches reduction in each successive pair of fixed and travelling sheaves, $1\frac{1}{2}$-inch distance rope centres is maintained, and the ropes finally anchor at the bottom of the fixed sheaves' bars. These multiplying sheaves are all mounted on roller bearings made of $\frac{1}{2}$-inch solid hard-steel rolls held in machined bronze cages made of one casting; they roll on hardened and ground steel trunnions and hardened and ground steel bushings $\frac{5}{8}$-inch thick. Grease is forced through the centre of trunnions into each sheave bearing. There are twenty-eight (28) of these $\frac{1}{2}$-inch rolls $3\frac{1}{2}$ inches long in each sheave. The load on these bearings is about 320 pounds per projected square-inch area, and their inaccessibility renders it necessary to make them able to run without attention for long periods of time. I have seen sheaves carrying this same load, but running their bored hole directly on the steel shaft, enlarge from 4-inch to $5\frac{3}{8}$-inch bore and wear the shaft $\frac{1}{2}$ inch where they turned on it, in six months.

The counterweighting effect of all these parts moving one-sixteenth of the car speed reduces the momentum of the total counterbalance, and, the vertical machine ropes not whipping as much as the horizontal machine ropes, produces a much better motion of car. These cars with their finished cages, averaging 40 square feet area of platform, weigh only 2,700 pounds, and are counterbalanced just enough to be operated, by the ropes, sheaves, and vertically moving parts of the machine, without a pound of metal being added for counterbalance only. The varying counterbalancing effect of the ropes as the car travels is compensated by two chains of $1\frac{3}{8}$-inch iron links looping from the counterbalance to a support in the building midway in its travel.

The vertical screw runs much truer than the horizontal and is much easier on the ball-bearing nut.

Our experience begins to indicate that our present size of nut, carrying 240 ½-inch balls under load at a time, is too small where they are geared 16 to 1 to the car and the load ranges from 100 to 300 pounds per ball. Fig. 300 shows a ball-bearing nut on a hollow screw bar carrying 320 ⅝-inch balls under load, which I hope to use in future high-geared elevators, thus getting the benefit of a stiffer and more reliable screw bar, as well as about three times the ball-carrying capacity. Mr. H. F. J. Porter, of the Bethlehem Iron Company, has offered to undertake to

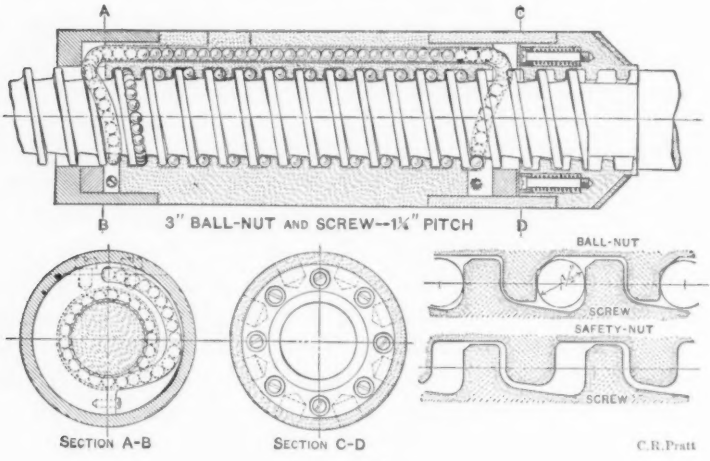


FIG. 299.

bore a 3-inch hole through the centre of these bars up to 26 feet in length, which, as this steel is 80 carbon and 60 manganese, is no easy feat to perform. This may be modified, however, by using a softer steel for boring and obtaining the requisite hardness afterwards by oil tempering.

Mr. G. H. Hill, E.E.—Early in its development as a commercial power machine the electric motor was applied to hoisting machinery. The first application to elevators for passenger or freight service was by belting to existing hoists formerly operated direct by steam or from a line of shafting. The motor in such cases is entirely similar to any power motor, as it is run in one direction continuously, and not started, stopped, and controlled by the car operator. The elevator motor, as a some-

what distinct class, is confined to those direct connected to the elevating machinery and controlled entirely from the car; the operations of starting, stopping, regulation of speed, and reversing being entirely functions of the motor and its controlling apparatus.

The necessity of meeting the severe conditions of these operations has developed a class of motors peculiar in some respects and with many admirable qualities. The conditions to meet are in some respects similar to those of street-car service, which has produced a type of motor of marked individuality and of high

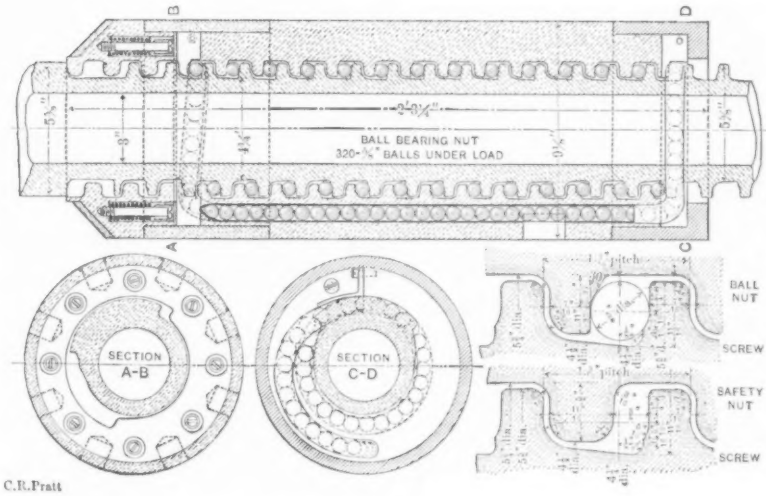


FIG. 300.

merit. The points of similarity between the two services—viz., elevator and street-railway—are chiefly thoroughness and strength of design, ability to stand all sorts of abuse, weather conditions, overload, shocks of frequent starting under heavy loads, and ability to reverse and run in either direction equally well. In several respects the conditions for elevator service are more severe and exacting than those for railway service, which seems natural, since an elevator is no more or less than a railway operation on a vertical plane.

Besides the prime requisite of a railway motor, viz., endurance—a requisite, by the way, much magnified in importance when we consider the impossibility of taking a disabled elevator home

for repairs while another replaces it—the elevator motor must be efficient and economical of power both on starting and running; it must be capable of rapid control, combining gentleness with almost unlimited power. An elevator is usually operated on a lower voltage supply than that of a railway line with corresponding greater current values. Frequently they are connected to circuits that supply lights and other power, and the number of machines in a single plant is so small as to prevent the averaging of the current value to a fairly constant figure, as is the case when a large number of units are in operation. The fluctuations of current are necessarily great, and, if excessive in starting, are especially noticeable and annoying. Any unevenness of action in starting and stopping is very noticeable and unpleasant. Their perfect safety must be beyond question.

Under the same conditions of field strength and potential the motor must have a very constant speed under extreme conditions of load from a negative to a maximum positive.

The most common and widely known form of electric elevator is the worm gear and drum machine, which consists generically of the winding drum and worm wheel carried on the same shaft, a worm meshing with the gear, and the armature of the motor mounted on the worm shaft. While not essentially so, these machines are almost exclusively operated on the overbalanced principle. This has the well-known advantage in economy of power by making the system balanced when the average load is in the car. With the car thus overbalanced the possibility exists—arbitrarily as far as the motor or control is concerned—of having the motor drive a positive load or of having the load tend to drive the motor in either direction according as the live load is greater or less than the amount of overbalance. This condition makes a shunt winding for the motor a necessity, since a series-wound motor would be useless to control the car where the load was tending to move by gravity in the same direction that the motor is being driven. When this condition exists with a properly designed shunt motor, a very small increase in speed over the normal, caused by the driving load, will raise the counter E. M. F. of the motor above that of the power circuit and cause it to generate current back into the line, thus acting as a dynamic brake.

A variation in speed of less than 10 per cent. is obtained with the best machines for a variation in load from the maximum

positive power of the machine to a similar negative load. This ability to generate current back into the line is of great importance not only in preserving a constant speed under all conditions, but also in the matter of economy. Where a number of machines are connected to the same power line close together, this action produces a remarkable saving in total power required. A remarkable instance of this is presented in a case where twenty-three machines, nearly all of the same size, are installed in one plant. The mains are of heavy copper and installed so as to closely tie all of the motors together. The machines are designed to carry loads of 5,500 pounds and 4,500 pounds at a speed of 150 feet and 225 feet per minute respectively. The regular average duty is very much higher than is ordinary—being usually about one-half of these load figures. The average hoisting current for each machine at 230 volts is about 50 amperes. The cars are rather heavily overbalanced, so that an empty car going up or a loaded car descending will drive the motor and cause it to generate current back into the mains, which goes to supply one of its neighbors that is running down light or up with a heavy load. The current supplied to the system is, therefore, the difference between the generated current and that expended in doing actual work. The result is a great diminution in power required by the plant over that which would be necessary were the elevators separated and supplied individually. The power required to operate the entire plant averages about 10 amperes for each elevator in service, and the total current supply is very constant. Instead of using a separate generator for the elevator load as originally contemplated, the elevators are now run from the same generators with the light load with perfect satisfaction.

In addition to having the motor design such that it will act as an efficient generator and attain a potential greater than the line voltage at a speed not greatly in excess of its motor speed, it is also a prime requisite that the worm gear be very efficient.

Another noticeable case of the effect of this action was mentioned in the discussion of an elevator plant at a recent convention of this Society. In this case two of the elevators of a plant of eight were used to carry load down only, the up trip being with empty car. The average current for these two machines is negative ; *i.e.*, they act as generators and actually supply energy to the other machines of the system.

The combined necessities of using a shunt field winding and—for reasons of economy—of having the current cut off from the field when the machine is not in actual operation, require a special consideration, especially in isolated machines, in order to prevent excessive inrush of current when starting. It is necessary to make the field magnetize quickly, so that the starting resistance may be cut out promptly. This is accomplished in some machines by a special arrangement of the shunt field, and in others by using a few series turns. These series turns must be cut out with absolute certainty as soon as the motor attains its speed, for the reasons before given.

For the purpose of slow running as well as to further reduce the starting current, the best practice designs the motor to run when at normal maximum speed with a field magnetization considerably below the saturation point.

When starting, the field is brought quickly to saturation, giving the motor a heavy torque and a slow speed with little more than the running current when the field is weakened and the motor speeded up. Such arrangement reduces the amount of power that can be obtained from a given motor, or, conversely, makes a larger and heavier motor necessary for a given output, as compared with standard power motors.

With the special conditions to meet, the best elevator motors have a very high efficiency. A hoisting efficiency over all—*i.e.*, from line supply to net load lifted by machine—of 60 per cent. is ordinary, while in many machines 70 to 75 per cent. is attained. For this the efficiency of the motor alone must be high, and is commonly 90 per cent.

The control of the drum type of machine has to perform three general functions:

1. Make and break the current and reverse the motor.
2. Cut out the starting resistance.
3. Circuit the motor through a resistance when stopping and apply the brake.

The oldest form of control accomplished the first and second entirely by a hand rope running to the car. These early machines did not accomplish the third-named, but depended entirely upon the brake for stopping. The motor affords a most excellent means for bringing the moving car to rest by circuiting through a resistance, preferably one variable from maximum to zero with heavy cars and loads. The action of the

motor is proportional to the load and speed and is powerful and at the same time gentle and easy. The action of a mechanical brake set at a given pressure is apt to be harsh with light loads, or not sufficiently powerful at heavy ones. With light loads and small machines this is not so noticeable.

The advantages to be gained by performing these functions, either altogether electrically or partially, have caused such means to be quite generally adopted. A well-known form of controller is one in which the making, breaking, and reversing are performed by a switch controlled directly by the hand rope, the same switch controlling the circuits of the brake coil and a solenoid which cuts out the starting resistance. The electric form of brake is in all cases arranged so that the brakeshoes are pressed against the brake pulley by a spring or weight, and released therefrom by a solenoid. This arrangement has the advantage of automatically applying the brake whenever the current is cut off in stopping or in an emergency. One form of solenoid for cutting out the starting resistance is wound so that its axis is a segment of the circumference of a circle. The plunger is similarly shaped and is carried by a lever on the opposite end of which is carried the brush which passes over a commutator between the segments of which is connected the starting resistance. An air dash-pot is usually attached to the lever to control the rapidity of movement of the plunger and lever. The lever movement is used because of the apparent difficulty of designing a solenoid magnet which will act over long distances. A form of controller manufactured by Sprague Electric Company which overcomes this difficulty is shown in Fig. 301. This is entirely electric—the making, breaking, reversing, and circuiting through a resistance when stopping being accomplished by the two solenoids on either side of the main solenoid. The rheostat solenoid in this controller is direct lifting against gravity and is governed by an adjustable air dash-pot in the interior of the coil. The contact brushes are carried on the end of the plunger, making contact with the rows of segments on either side. The movement of this plunger is about 10 inches, and its rapidity of movement is easily varied by the air dash-pot from 1 to 5 seconds as desired. Another form of entirely electric controller is the magnet control manufactured by the Otis Company. This controls the making, braking, reversing, and cir-

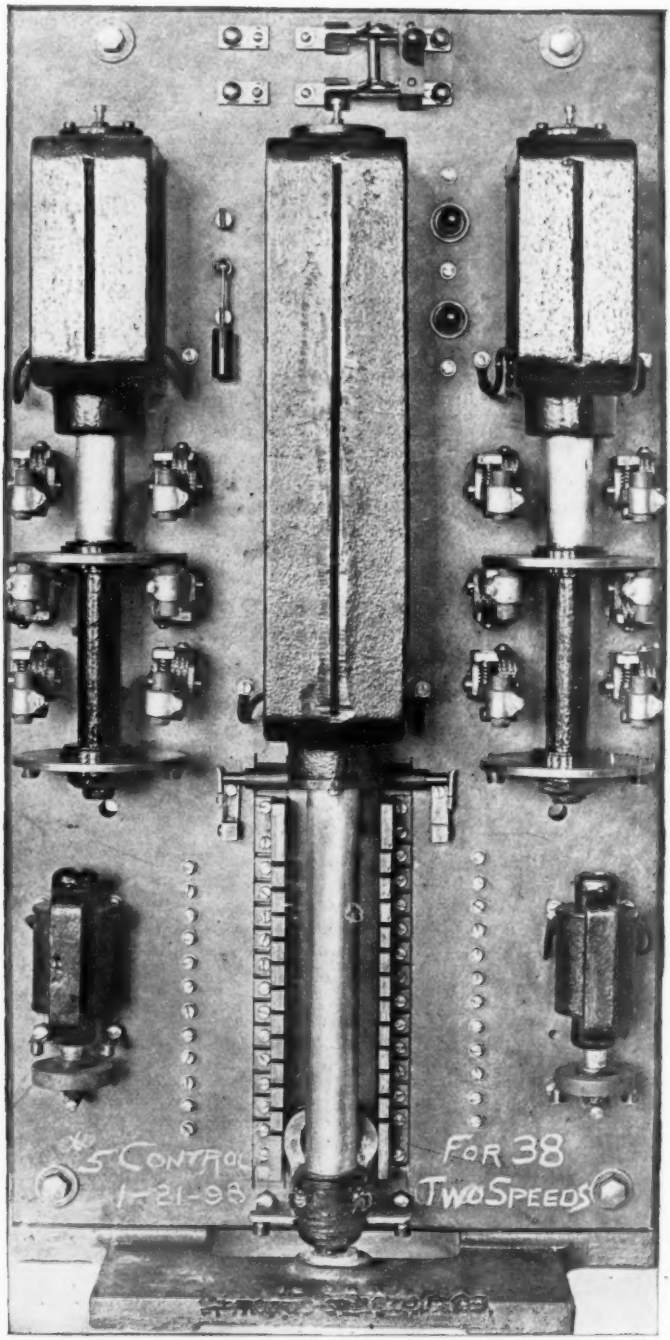


FIG. 301.

cuiting, by magnets in a way similar to the above. The starting resistance is cut out in sections by the action of a number of other magnets whose coils are so wound and connected that they are energized in rotation as the armature of the main motor increases its counter-electromotive force. The potential thrown across the armature terminals is thus proportional to the counter-electromotive force it has attained, the acceleration on starting being automatically proportioned to the load. On this and the above controller a variation of speed of the motor is accomplished by a separate magnet which varies the field strength of the main motor.

In all of the above controllers the cutting out of starting resistance, and the speed with which this is done, are independent of the operator in the car; the variation in speed under the operator's control being entirely that produced by field variation of the main motor.

The pilot-motor controller manufactured by the Sprague Company provides means for controlling this at will. This is shown in Fig. 302. The making, breaking, and reversing are controlled by the two circuit breakers as in the other forms. The control of the starting rheostat and field variation is through the medium of a pilot motor the circuits of which are controlled by the operator. The pilot motor is series wound with differential field, and is started and reversed without any starting rheostat. It is provided with limit switches and stops by which it cuts off its own circuit at the limits of its motion. It is also provided with a brake similar in construction to the main-motor brake and mounted upon the armature shaft. The coil of the brake is connected in series with the motor. This brake serves to stop the movement of the rheostat arm instantly, when the current is cut off from the pilot motor by the car operator or at the limits of its motion. With this form of controller, the speed of the car is under complete control from the very slow movement with full field strength and all starting resistance in circuit, to the maximum speed with the motor across the line and the field weakened.

An interesting device is provided on this control for making the acceleration of the car when starting proportional to its load. The device consists of a solenoid in circuit with the main-motor armature and provided with a plunger and suitable contacts. When the starting current exceeds a predetermined

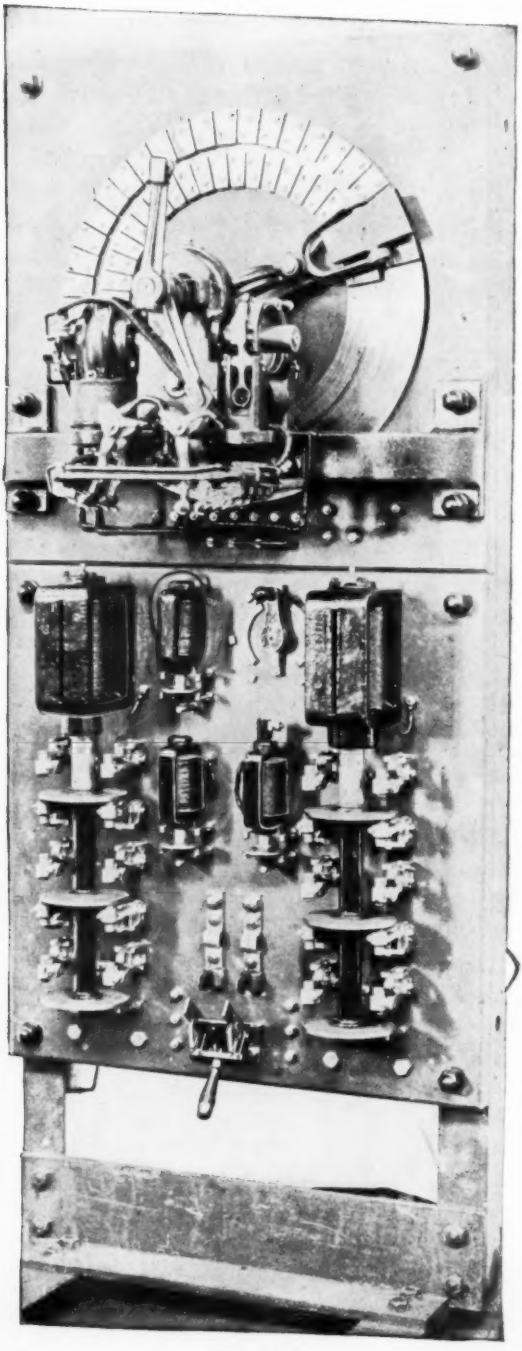


FIG. 302.

amount for which the throttle (as the device is termed) is set, the plunger of the magnet draws up, closing a shunt circuit about the pilot-motor armature. This slows down or stops altogether the movement of the rheostat arm until the counter-electromotive force of the main motor increases and the current again falls below the given amount, when the plunger drops and the movement of the rheostat arm continues.

With the complete electric control there is usually provided a slack cable switch operated by various devices, which cuts off the current and stops the motor in case the hoisting cables become slackened in an emergency, and a centrifugal governor which acts similarily in case the motor attains too high a speed.

Another form of control for drum elevators, quite different from all the above, is that known as the Leonard system. The peculiarity of this system lies, not in the motor, but in the applied power. By means of an individual generator for each elevator, whose field strength and therefore voltage are varied or reversed at will by the operator, this supplies to the motor, without the use of resistance, the variable voltage for starting, as well as for speed regulation when running. The field of the motor is kept at constant strength and can be maintained at a maximum.

The starting power with this system of operation need never exceed that required for running, and in this respect, as well as in the directness and simplicity of the idea, it is in theory the ideal method of control. It is, however, hampered commercially by the necessity of the individual and separate generating apparatus. Special consideration must be given to the construction of the generator, in order to have its field strength and voltage follow quickly the action of the operator in increasing or reducing the field current, in order to produce rapid control of the car. One of the strongest advantages of the electric elevator is the possibility it presents for centralizing all the power needed for a building in a few generating units. Where the generators cannot be used for other purposes or even interchangeably with the lighting or other power generators, the electric elevator loses a great part of its advantage over other systems.

Another type of electric elevator which has attained prominence is the screw and ball-nut type manufactured by the Sprague Electric Company. This is a radical departure in its

electrical as well as mechanical features from all other electric-elevator practice, and was designed to duplicate hydraulic-elevator service in the matter of speed and operation for high-duty passenger service in office buildings The machine has been described above.

The car in this system is always underbalanced. The motor operates as such when hoisting, but is driven as a generator by the load when descending. The motor is series wound with a few turns of shunt winding remaining in circuit to assist the rapid building up of the field magnetization when starting. The form of control for this machine is shown in Fig. 303.

The making and breaking of current (there is no reversal of current or motor connections, since power is used only in hoisting) are performed by a circuit breaker similar to that for drum machines. The speed in hoisting or lowering is variable at will over the entire range of the motor, and is controlled through the medium of a pilot motor which is geared to the rheostat arm. On the hoisting motion this is an ordinary rheostat control; in lowering, the motor as a generator is circuited through a variable rheostat, the retarding effort, and hence the speed, being governed by the amount of resistance in circuit.

The brake is connected direct on the armature shaft, and is similar in principle to that for drum machines.

The positive connection between the power unit and the car, the positive limits of travel presented by the ball-nut and screw system, and its great adaptability to the application of safety devices, give this type of elevator marked advantages over the drum machine. The efficiency of the machine is high, 70 per cent. hoisting efficiency from power supply to load hoisted being ordinarily attained.

The introduction into extensive commercial use of two-phase systems of alternating-current electric power, and the use of this as well as single- and three-phase systems abroad, have produced a demand for elevators that will operate on such systems. A number of such are now upon the market. The motors for these are usually standard types, as built by the General Electric, Westinghouse, and other companies.

The controlling apparatus is much simplified by the lack of necessity for any starting resistance. The low efficiency of alternating current magnets requires that special consideration be given the control apparatus in order to prevent too much power

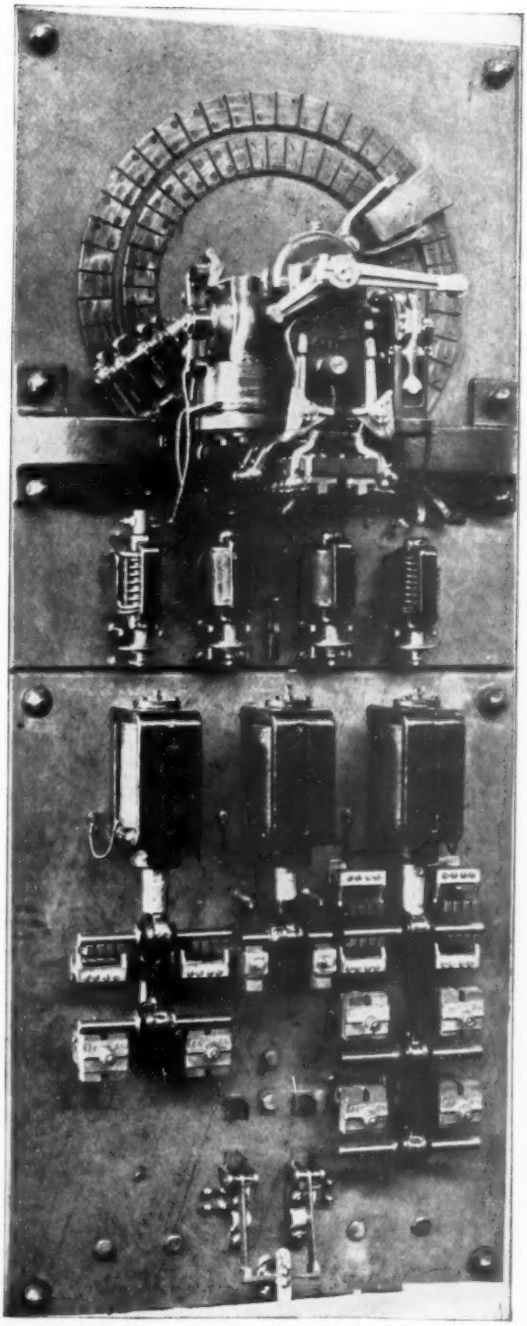


FIG. 303.

being expended therein. In one type of alternating machine a small motor is used to operate the circuits of the main motor and to lift and apply the brake. The objection is open to this, that in case of line failure the brake would not be applied or the circuits of the motor opened. Another simpler and more direct apparatus controls the reversing switch of the motor, and operates the brake by means of a solenoid magnet. When the plunger of this magnet has completed its movement and at the same moment that current is applied to the motor, it opens its own circuit and is caught and held by a latch. Current is not flowing, therefore, through the solenoid and motor at the same time, a point of advantage in keeping the starting current low. To stop the motor and apply the brake the latch must be released by a second small magnet. At the limits of motion of the car this latch is also released by mechanical means. This is partly open to the same objection as the first-named, as the brake would not be applied nor the circuits opened in case of failure of power for a short or long period, except when the car reaches its limit of travel.

Another form not open to this objection employs two solenoids for lifting the brake and reversing the motor, one for up and one for down motion. The solenoids are so designed that when the movement of the plunger is completed the reactance of the coil is greatly increased, and hence the current diminished in value much below that required to raise the plungers. This form of control stops the motor and applies the brake instantly when the control circuit is opened in normal operation, or in an emergency. In case of single-phase motors a teaser arrangement is employed to produce a rotary field when starting, and is cut out when the motor reaches synchronism with the line. All forms of electric elevator mechanism lend themselves very readily to the application of special and automatic devices for safe and convenient operation and to accomplish any special object. The forms of such special application are of great variety, and a detailed description of them would be quite beyond the limits of this paper.

A very popular and interesting form of automatically controlled elevator for use in private dwellings deserves mention, however. The machine itself is of the ordinary drum type, having a moderate speed, usually about 100 feet per minute. It is designed to be operated by the passenger, no operator being

required. Each floor landing is provided with a button switch, by pressing which the car is brought to that floor irrespective of its previous position, provided it is not already in service

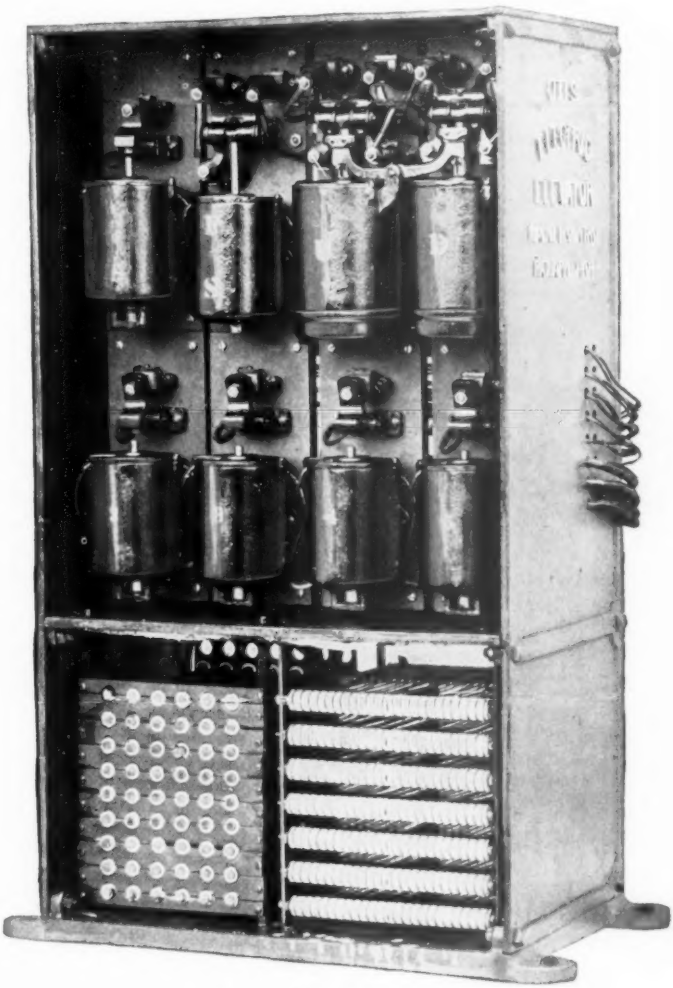


FIG. 304.

and all landing doors are closed. Each landing door is equipped with an automatic lock, released by the car when it is opposite the landing, and holding the door closed at all other times. It also is provided with a safety switch, which opens the control-

ling circuit whenever the door is opened, so that the car cannot be operated unless all doors are closed. The car may be provided with several forms of operating devices, all of which are so arranged that when being operated by a passenger the landing buttons are inoperative. These types of elevators are in extensive use, and have shown themselves not only perfectly safe, but very convenient and reliable.

The same, or similar, form of automatic control is used in more or less elaborate modifications for a great variety of purposes, such as dumb-waiters, parcel hoists, freight service, etc., etc., which the flexibility of the electric control makes possible with a simplicity and directness quite impossible with any but electric machines. The descriptions of the various types of controls given have necessarily been of the briefest possible character. There is much of interest to be said in description of details and in discussion of the merits and demerits of the various types of machines and controls as well as upon the questions of power, economy, and efficiency. Any attempt at this, however, is quite impossible in a single paper.

DRAWINGS AND PHOTOGRAPHS FOR ELEVATOR PAPER.

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Rack and Pinion Inertia Car Safety	272
Baldwin Corrugated Roll Car Safety	273
Ellithorpe Toggle-joint Car Safety	274
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DISCUSSION.

Mr. W. S. Rogers.—It is interesting to read that some one connected with the electric elevator business considers a ball-bearing nut to be a crude affair as a ball bearing. My experience has led me to the conclusion that it would have been wise to have listened to the advice of Mr. C. A. Lieb, quoted on page 837, that it was a curiosity of no practical value.

We are told that the ball nut is "harveyized" in bone dust and charcoal. Why not say that it was "case-hardened," as that is all there is to it, and it is a process which we have all been using at various times for many years, and know that it is not always a "sure thing," as the best steel workers will testify.

Now, referring to Fig. 299, we notice that the walls of the area in which the balls roll form a square, with the exception of the corner of the nut against which the thrust load comes, which is left with a fillet of 30 degrees. When at work the lead of the ropes is parallel to the line of the screw, while tangential inertia of the balls is at right angle to the screw. These two forces produce a resultant force compelling the balls to wear away the 30-degree fillet and make for themselves a true rolling path. While this action is going on there is also a tendency of the balls to wedge outward until they have cut the angular portion of the nut into slivers and splinters, giving themselves room to work in. (I will state that I have seen these splinters of steel an inch long and $\frac{1}{32}$ of an inch thick.) These splinters and cuttings also are gathered in among the balls in a compound of metal succotash, which assists in making some balls larger than others, causing them for the moment to take more of the horizontal load than they are able to carry, and the result is broken balls, elevators stopped for repairs (and generally during the busiest hours of the day), and patrons and owners all out of humor, while the poor mechanic in the engine room has to stand the brunt of all the trouble. Then the so-called expert is called in, and he immediately begins an examination of the broken balls with a microscope, endeavoring to find the microbe which caused the fracture. The second element helping onward the destruction of the mechanism lies in the fact that the thread of the nut is not a *perfect* pitch, because such things are impossible, and the same holds good with the threads of the screw, and in conjunction with them we know

that the best steel balls are not all of the same exactness, perfect in gauge and spherically, and practically not half of the 240 balls are under load at once. Knowing this to be true, and that a ½-inch ball's maximum running load is but 200 pounds, it is easy to see that the nut is only able to resist 24,000 pounds. Nevertheless, we are informed on page 844 that the maximum average load on the thrust bearing is 30,000 pounds, with occasional shocks of 80,000 pounds. With this discrepancy in design, is it any wonder that the weaker member of the construction continually gives out, even if there were no other destructive elements surrounding it? Lastly, the balls are of tempered tool steel with two axes of rotation. Consequently, in action they grind against the two walls of resistance, which, being much softer in texture, are bound to wear away, destroying any accuracy they may possess, thus allowing a few of the balls at times to take the entire load until such time as crystallization is complete. Then they break, and the expert goes after them with his microscope. In one instance, where balls made of inferior iron and hardened by the Simonds process were used, they wore down from half an inch to almost three-eighths of an inch in diameter without breaking, and there is no guessing as to how much longer they would have lasted. Another argument favoring the truth of this is the fact of the so-called "cold-rolling process" mentioned on page 844, where they perform in a laborious, mechanical manner a trick which any intelligent toolmaker can do in an hour's time—simply drawing the temper of the balls. In the meantime the experts are driving the ball manufacturers almost insane with propositions to "test" balls made of different kinds of steel.

Referring to the thrust bearing, Fig. 298, the rollers will stand safely against a load of 120,000 pounds. The balls used previously would only carry safely 36,000 pounds, while 80,000-pound shocks are stated on page 844 as likely to occur. In our business we do not recommend this type of bearing, unless there is absolutely no room for the ball type, as the friction is increased fully 50 per cent. I do not consider the roller bearings for the sheaves described on page 847 to be what they ought to be, or that the rollers will last as long as they should. In my opinion it is possible to design roller bearings which can be used for twenty years.

Mr. William H. Bryan.—The description of the Ellithorpe air cushion is particularly interesting, and I would like to know to

what extent it has actually come into use. It would seem to be the best possible safety device for a falling elevator. Is there any danger of "teetering" when the elevator strikes the cushion? If so, is it not possible to displace the cables, or to drop the counterweights? Figure 281 shows a batter at the bottom of the well, and there also seems to be several inches of clearance around the platform of the car at the narrowest place. Have any experiments been made to determine whether this space can be so adjusted as to control positively the air pressure? If so, it might be possible to do away with both the suction and relief valves, thus eliminating two features which might be tampered with or get out of order.

Our experience with guide rails does not correspond with Mr. Pratt's. We find there is usually a saving averaging about $100 per elevator for wooden rails over steel, even where the former are of the best possible construction. This is partly accounted for by the more expensive safeties required when steel guides are used.

Mr. Pratt brings out very clearly the advantages of the electric elevator in a number of important features:

1. The location of the machinery at top of hoistway economizes space and lessens friction by reducing the number of sheaves.

2. The car may be counterbalanced, not only for the dead load, but over-counterbalanced to the average live load, which arrangement gives the highest possible motor efficiency, permits the use of a smaller motor, and tends to equalize the current consumption.

3. It greatly simplifies and cheapens the power plant.

The improved efficiency of the worm gear is interesting, as is also the fact that it has now been shown that a single worm gear with roller thrust is more efficient than the double worm.

Mr. Pratt states that accumulators are chiefly used with high-pressure hydraulic plants, but that pressures above 250 pounds per square inch are uncommon. There are a number of hydraulic plants in St. Louis operating with pressures of about 800 pounds whose performance is satisfactory and whose fuel economy is high.

Mr. Pratt is probably correct in stating that the machine shown in Fig. 291 was the first electric elevator ever built, if he limits it to direct-connected machines; but electricity had been used for running elevators earlier than this, both by motor-driven

pumps for hydraulic elevators and by motors belted to power elevators.

The screw machine described is admirably adapted for high lifts and speeds, and is, in fact, the only type of electric elevator which can be used for this class of service, but on account of its large starting current, and the fact that it cannot be over-counterbalanced, and returns no current to the line, its power consumption is always high compared with the drum machine, and larger generating units must be supplied. I suggest that Mr. Pratt add to the paper some characteristic curves of current consumption of a typical machine of the latest screw type for different loads.

The hoisting efficiency of 70 per cent. given on page 845 is probably for gross and not live load. If measured on the basis of watt-hours per ton-mile of live load only, and including starting current, and taking readings for the entire round trip, the result would be quite different.

Mr. Hill's supplement to the paper is interesting, in that it calls attention to recent developments in motors by which the current consumption is reduced and the surge at starting entirely avoided. A number of manufacturers are now offering elevator motors under guarantees that the accelerating current shall not exceed the running current, and this without making the acceleration slower, a result which they claim to accomplish by special winding. It means a somewhat larger and more expensive motor than would ordinarily be employed, and one whose efficiency is a trifle less after it is once under way, but the increased expense of the motor is more than met by the reduced size of the generating apparatus. The absence of excessive starting current brings up the gross or round trip efficiency, particularly where the elevators make frequent stops. It is now the almost invariable custom to run electric elevators from electric lighting generators, and when properly adjusted both classes of service are entirely satisfactory.

While the drum electric machine has many advantages, it should not be operated at higher speeds than 350 feet per minute; and if the lift is over about 150 feet drum complications are introduced. The most successful drum machines are built to operate at practically constant speed in both directions, irrespective of the load. In freight service it sometimes becomes necessary to handle loads of, say, double the rated capacity of the elevator. On these rare occasions it would be permissible to reduce the speed to one-half or even less. If the elevator people

would develop a motor which would fill this requirement without being unnecessarily complicated or expensive, and while maintaining proper efficiency in regular service, I believe it would prove popular. In this connection it is interesting to note the development in St. Louis of a single-phase alternating motor which is self-starting, and which may prove itself well adapted to elevator service in those cities where only single-phase alternating current is available. As at present constructed, it sparks during starting, and is, therefore, better adapted for continuous running, being belted by tight and loose pulleys to power machines.

The important advantage which the drum type of elevator has—of returning current to the line—has already been mentioned. This may be done either when a heavily loaded car descends or a lightly loaded car ascends. This introduces an interesting question. What happens when one or more elevators are returning current to the line at a time when the output of the generator is a minimum? It is not difficult to imagine all the elevators stopped, except one or two. If at this time the output of the generator for lighting is small—which might occur in the daytime in a building using but little artificial light—it is possible that the return current might not only exceed that used for the lighting and power at the moment, but also that necessary to run the dynamo as a motor. Under these conditions, the generator and its engine would increase in speed, which would result in the engine governor, if of the proper type, cutting off all the steam. What would happen next? The elevator motors, which are now dynamos generating current, have, of course, only a limited driving power behind them. In most cases this would not be sufficient to more than drive the dynamo as a motor and the engine connected to it with but little increase of speed. Assuming, however, that the return current might exceed even this, what would happen with a compound-wound generator such as is always used for this class of work? The return current converts the dynamo into a differentially wound motor—that is, one in which the magnetism due to the current in the series winding is opposed to that due to the current in the shunt winding. The rotation of this differentially wound motor will be in the same direction as when it ran as a dynamo. Its speed will depend entirely upon the relative proportions of the series and shunt windings. If the dynamo is heavily compounded, its speed as a motor will increase

as the current increases. The standard generator is wound for 10 per cent. overcompounding, and this is reduced by the shunt compounding rectifier to about 3 per cent. for isolated lighting. Under these conditions the speed of the motor will accelerate but slightly. The probabilities, therefore, in such a case are that the dynamo would operate satisfactorily and without material increase of speed until a considerable quantity of current should be forced through it.

The remedy—if indeed a remedy is necessary—is to place between the dynamo and switchboard, not only the usual overload circuit breaker, but also an under-load circuit breaker, which will open when the current falls to zero. A better arrangement will be to use a circuit breaker which would pass enough return current to run the dynamo and engine at normal speed, but would open for any greater current. No such instrument has been developed, however, so far as I am aware. It is also important in such a case that the governor of the engine be of such a type as to entirely shut off the steam at a given excess of speed.

The problem above presented has no doubt been thoroughly considered by designers of generators and motors, and it would be interesting to learn their views. In view of the fact that thousands of electric elevators are now in use in this country, and that many of them return current to the line, it would seem that this condition must frequently have arisen. There seems to be no record, however, of an accident from this source, which would indicate that it is impossible for dangerous conditions to arise.

*Mr. Chas. R. Pratt.**—On pages 825 and 826 reference to the liability of drum elevators running the car or the counterbalance into the top of the hoistway was intended more as a comparison with the other types mentioned, in which this could not possibly occur, rather than to imply any serious and actual danger pertaining to the drum type.

On page 827 my remarks on Hindley worm gears are a warning to manufacturers not to attempt to build them in small quantities for light duty where there is space to use a plain worm gear. The purchaser, however, should give the Hindley gear a preference, where it is properly built, as it will be more efficient and durable for any purpose he may choose to use it.

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* Author's closure, under the Rules.

On page 844, in regard to the cylindrical roller bearing shown on page 845, Fig. 298, I should have stated further that there are now about 200 of these taking the thrust of the screws on Sprague-Pratt electric elevators, and not one of them has as yet shown any measurable amount of wear or other deterioration.

On page 846, the statement that the safety nut is intended to hold the load in case the thread of the ball-bearing nut wears out and strips is not intended to imply that this ever occurs. It certainly might occur in time if the engineer never cared for or inspected the ball-bearing nut, but it has not occurred yet, and probably never will occur, but this improbability does not do away with the advisability of having such an absolute safety device on a passenger elevator.

The ball-bearing nut needs no defence from Mr. Rogers's attack, but, as a matter of record for our *Transactions*, it should be stated that this nut was never expected to be an ideal ball bearing as compared with ball-bearing journals or thrusts, where the balls roll upon perfectly true and glass-hard surfaces and carry very little load per ball. Such bearings show no appreciable deterioration in a lifetime when properly made. This nut was made for the specific purpose of operating an elevator car with a speed and motion equal to the best hydraulic elevator, at no greater first cost or cost of operation, and with all the natural advantages of electric power over hydraulic incidental to the arrangement of the power plant in large buildings.

The larger size of this nut shown in Fig. 300 has been made and tried since this paper was presented at Washington, and tests of its durability indicate that it will last as long as any piece of high-speed elevator mechanism is expected to last.

The comparative degree of success that this nut has attained commercially is not an argument for our Society. But if it was as Mr. Rogers describes it, it could scarcely have been selected as the means of operating elevators in the finest and highest buildings in nearly every large city in the United States for the last six years. The latest installation of the ten passenger elevators in the Park Row Building, New York City, made the trial run of thirty days without an instant's interruption to the service of any elevator for any cause whatever.

Mr. Rogers's remarks on the Harvey process and upon case-hardening would lead one to suppose that they were one and the same thing and equally ineffective for practical results. As the limits of this discussion will not permit of a complete description of these methods of hardening steel, the author can only call attention to the fact that the armor plate of every war vessel afloat is Harveyized, and the bearings of small arms, bicycles, sewing machines, etc., are nearly all case-hardened. Therefore, neither of these processes can be fairly considered to be either obsolete or experimental, as would be inferred from Mr. Rogers's statement.

Mr. Rogers next attempts to explain various causes for undue wear and tear of this ball-bearing nut, suggesting "tangential inertia" and other equally improbable reasons for the same. The one only reason why some of these nuts, balls, and screws have not lasted as long as they should, is because they have been overloaded. This fault is now recognized by their manufacturers, and the remedy, in the form of the larger size, as already described in this paper, is now being applied to such plants as may require it, and the members of this Society will be invited to inspect the first installation of the same at the Waldorf-Astoria Hotel, during our next convention in December, where they can also observe the smaller size working under an overload.

The author is perfectly familiar with the process of drawing the temper of steel balls, and most of the balls used in this nut have had their temper drawn to whatever temperature was best suited to the steel used, but it does not produce the same result as the cold rolling action that these balls receive under their regular duty.

Mr. Rogers's remarks about the inadequacy of the roller sheave bearings described on page 847 are totally uncalled for, as these bearings follow the best practice in this line, and are more substantially built than any of the regular commercial types of roller bearings.

Mr. Rogers's discussion concludes with an attack on the efficiency of the cylindrical roller thrust bearings. While the efficiency of this cylindrical roller thrust bearing is something over 95 per cent., and higher than any form of ball or cone-roller bearing ever used in its place, it is easy to see how the efficiency of this form of thrust bearing might be reduced to

almost any extent by increasing the length of the rolls in proportion to their radius of rotation.

Replying to Mr. William H. Bryan's discussion: First, in regard to the Ellithorpe air cushion, in Mr. Ellithorpe's list of references are 276 elevators in the finest buildings in most every large city in the country. The relief valve performs a function impossible to produce by any graduation of clearance between the car and the enclosed hoistway where the car falls from varying heights with varying loads, wherein this valve is adjusted to bring the car to a stop in a distance of such proportion to its free fall as to avoid accumulating sufficient pressure to cause the car to rebound and "teeter." Neither can any graduation of clearance between car and hoistway serve the function of the suction valve, which materially reduces the partial vacuum caused by the car being hoisted out of the air cushion at high speed.

As neither of these valves depends upon tight joints, packing, or any kind of close fits, there should be no possibility of their getting out of order.

In the matter of wood *versus* steel guide rails, where the hoistway affords solid supports near enough together to use a plain wood post to fasten the wood guide rails to, a first-class job can be erected for less cost than steel rails, even when the posts are made of built-up plants of kiln-dried white pine, which is the best timber for this purpose. But when the building only affords supports at each floor the usual construction is to build the wood posts in between the backs of two channel irons, and this costs more than steel rails.

In regard to comparative efficiencies of single and double worm gears, I did not intend to convey the idea that the single worm is, or could be, very much more efficient than the double worm gear; the point I wish to emphasize is that, as it is possible to get as much or greater efficiency out of a single worm gear as out of a double, there is no object in using the latter, which is the larger and more costly machine of the two.

The author is aware that there are several successful hydraulic elevator plants in St. Louis using pressures of about 800 pounds, but they are for freight duty, and plants of this pressure for high-speed passenger service have not been as successful or extensively used in other cities.

The difference between live and gross loads on Sprague-Pratt electric elevators varies very much with the condition of the buildings where they are used, on account of light or heavy cars; heavy cars cannot be operated with the same amount of unbalanced weight as light cars, and the different requirements for starting and stopping time also affect the amount of unbalanced car, so that machines exactly alike in every particular would vary in kilowatts per car-mile under equal live loads in different buildings to a great extent. Therefore, the characteristic curves of current consumption would not be instructive unless given for several plants covering all typical conditions, which would exceed the limits of this paper.

Mr. G. H. Hill.—Mr. W. H. Bryan refers at some length to the probable results of certain possible, though hardly probable, conditions of an elevator installation where the load is driving the motor as a generator. The writer has never known of such a case in a commercial plant. Where the elevators are connected to the street mains, it is, of course, quite impossible, and in case of a local plant it is quite improbable that the generator would have a load so light that it would not take care of the return current from one or two elevators running under the conditions assumed without having any effect upon the generator itself. Even if the generator were without other load than that of the elevators, the effect would be not enough to more than speed the generator (as a motor) a little above its normal, the power required to drive the engine being sufficient to absorb that delivered by the elevator. As the line resistance in such a case would be greater than usual, the elevator would have to acquire a higher potential than usual, so that its speed will not be kept so constant. Just such an action as Mr. Bryan speaks of did actually take place at a test of one of the Sprague Electric Company's type "S" elevators for the Central London Railway at London. These machines are built for a load of 10,000 pounds net and a speed of 200 feet per minute. When the first machine was erected for trial a small direct-connected unit was installed solely for the purpose of supplying current for the test. When the car was allowed to descend with a full load, the current returned to the dynamo was sufficient to run the dynamo as a motor and its engine at a speed that created some surprise. The installation of an under-load circuit breaker served to prevent the recurrence of this incident.

DCCCXXI.*

THE EQUIPMENT OF TALL OFFICE BUILDINGS IN NEW YORK CITY.

BY REGINALD PELHAM BOLTON, NEW YORK CITY.

(Member of the Society.)

In a space in the lower part of New York City, one mile in length and two-thirds of a mile in average breadth, there are at present located some fifty-six buildings exceeding 180 feet in height, while a few of these range from 200 to 325 feet from the sidewalk level. The demand for office accommodation in this district began in the vicinity of the Stock Exchange, and has developed buildings up the line of Broadway and parallel streets as far north as Franklin Street and as far south as Bowling Green. The convenience of the various exchanges has grouped the different professions and contributing trades in certain districts, and the legal profession has been attracted to the district around the municipal buildings and the courts in City Hall Park.

While at first, when there were few such examples, they returned a fairly profitable investment without much regard to economy of operation, yet in recent years the construction of so many great buildings has brought about a sharp competition which has called attention to economical equipment and operation, and has directed the attention of engineers to the problems which they offer.

Cost of Land.

The cost of desirably located sites has been of recent years greatly on the increase, and in certain advantageous positions has reached as high a figure as $300 per square foot. The corner plot on which the German-American fifteen-story building stands, 111 feet on Liberty Street and 79 feet 2 inches on Nassau Street, cost $937,000, or about $110 per square foot, previous to the

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projection of the new building, and was later on purchased for the site of the present tall building at $140 per square foot.

The land on which the Hudson (sixteen-story) building stands, facing 45 feet on lower Broadway, extending 225 feet to New Street, was purchased some years ago for an average rate of $65 per square foot, and would for the purpose of the present increased building be worth more than double that amount. An average value of sites for such buildings as are under discussion would now be $150 per square foot. A recent purchase of 1,736 square feet of ground by the Stock Exchange on Broad Street is at the rate of $244 per square foot, while an extreme was reached by the price of $330 per square foot paid in 1882 for a very desirable corner site in the same neighborhood.

Increased Returns.

Such large amounts as these values represent naturally call for a large return from the building occupying the site, such return being obtainable in the old five- and six-story buildings only by means of increased rentals. High rentals had, however, the reflex effect of driving tenants to less advantageous locations in rear streets until the height attainable in the modern buildings effected a combination of improved accommodation at a reasonable price. The effect of the above natural causes has been, therefore, even with a greatly enhanced cost of the site, a reduction in renting charges proportioned to the increase in rentable area. The competition between the various buildings has brought the minimum rate down from $3 per square foot to below $2 per square foot of office space, and includes in this charge light, heat, attendance, cleaning, and, of course, elevator service.

Relative Increase.

It may be roughly estimated that the increased height of the downtown buildings referred to has added to the inhabitable area not less than 7,500,000 square feet. Illustrating the relative increase of value obtained by height, the four- and five-story buildings occupying the site of the present German-American Building represented a renting area of not more than 6,000 square feet per floor, which at the old rates of about $3 per square foot per annum may have produced a gross rental of as much as $70,000 per annum when filled, but even this represented only 6 per cent.

on the real value of the land alone. The new building has a net renting area of about 7,000 square feet per floor, producing on fourteen upper floors, even at the low average of $1.60 per square foot, $156,000, in addition to which are two large ground-floor banking parlors producing about $17,000 and $12,000 per annum respectively, also a basement restaurant bringing in $3,000, or a gross return of $188,000 per annum, an increase of two and a half times as much as the old buildings. On a cost for building and equipment of $650,000, and of land $937,000, or $1,587,000 in all, a gross return of nearly 12 per cent. is thus attained. The cost of management, operation of equipment, and city taxes naturally form a large increase on the same items in old buildings, but under good management and economy do not exceed 50 per cent. of the gross rental. It will be evident that the economical operation of the equipment forms a decided factor in the net balance available for interest.

As a mortgage can be placed on such a building and land for six-tenths of its value at a rate not exceeding 4 per cent., the net return on the actual capital sunk in such a building may be very high; nevertheless wasteful appliances and accompanying cost of management reduce the return in many instances down to 3 per cent. Once such buildings with moderate rentals and greatly increased advantages are established, they form a strong incentive to the erection of other buildings of similar or superior character owing to gradual abandonment of the older buildings by tenants, in favor of those where they find better conditions. The sanitary appliances in the older buildings were and are extremely behind the times, and have in numerous cases been the cause of tenants' removal. The old three- and four-story buildings on the site of the Bowling Green Building, from the above causes, became unremunerative and some were closed up to avoid taxation. The site, which fronts 157 feet on Broadway, extends 170 feet through to Greenwich Street, and was valued at a million dollars by the building company. On this site the Bowling Green Building has been erected, which is the largest commercial office building in New York City, being 240 feet 6 inches average height from street levels, and of a gross content of 4,915,000 cubic feet, with a floor area of 20,555 square feet on fifteen upper floors, and of 32,000 square feet on the ground floor and first basement, the sub-basement being devoted to mechanical services and storage. In this instance the increase in acreage has been enormous, the

original 80,000 being raised to 567,500 square feet, or fully four times greater renting area than before. The total cost of the building was close on two million dollars, and owing to the remarkable economy of its equipment and management it has been able to pay its way when not more than five-eighths occupied.

Size of Buildings.

The best method of comparison of size is that afforded by bulk. Representative instances are the following:

The Bowling Green Offices, 16 stories above street, 235 feet high, 5,000,000 cubic feet gross contents.

New York Life Building—partly insurance—16 stories, 4,800,000 cubic feet.

Manhattan Life Building—partly insurance—17 stories and tower, 270 feet high, 1,665,000 cubic feet.

Lord's Court building—offices—220 feet high, 3,000,000 cubic feet.

American Surety Building—partly insurance—21 stories, 305 feet high, 2,176,000 cubic feet.

These represent either abnormal height or bulk, and of them the Bowling Green and Lord's Court buildings are those built on a strictly commercial basis.

The representative commercial office building now averages 200 feet high, and of these there are now a large number.

Average samples are:

The Central Bank Building, 16 stories, 1,980,000 cubic feet.

The German-American Building, 15 stories, 1,400,000 cubic feet.

The Hudson Building, 16 stories, 1,500,000 cubic feet.

R. G. Dun Building, 15 stories, 1,200,000 cubic feet.

The above buildings also afford an opportunity for interesting comparison on account of the dissimilarity of their elevator systems:

		Boilers.
1. Bowling Green Offices, 9 hydraulic elevators	720	horse-power
2. Lord's Court building, 5 electric elevators	420	" "
3. Central Bank Building, 5 hydraulic elevators	460	" "
4. German-American, 5 electric elevators	320	" "
5. Hudson Building, 4 hydraulic elevators	500	" "
6. R. G. Dun Building, 6 electric elevators	600	" "

All of the above have been equipped to the plans of the author, and are, with the exception of the last, heated by the Webster

system of vacuum returns with exhaust steam at atmospheric pressure.

The boiler capacity includes, in all but the last instance, a spare or reserve boiler. The R. G. Dun Building has a single pipe back-pressure system of steam heat, and has only two boilers, both being in use in extreme weather.

The services provided in each are:

1.	Car-miles per hour,	14 to 20	; lights,	6,200...	Bolton	vertical	system of	wiring
2.	"	"	9½	"	4,000...	"	"	"
3.	"	"	11	"	3,800...	"	"	"
4.	"	"	11	"	2,800...	Horizontal	wiring	
5.	"	"	8	"	3,000...	"	"	
6.	"	"	16	"	4,500...	"	"	

Total Cost.

The cost of average steel-framed 16-story office buildings complete with equipment, and inclusive of a moderate amount of decorative outside treatment in stone machine mouldings or in terra cotta, without hand stone-carving, is 36 to 40 cents per cubic foot of the gross cubic content, outside measurement. The excessively high buildings cost more, also those with expensive adornments. The relative cost of the mechanical appliances, including power, elevators, heat, light, and sanitation, is approximately one-seventh of above. The total cost of mechanical appliances in a building of 16 stories, basement, and sub-basement, say 6,000 square feet renting area per floor, was $82,000.

Or in detail:

	Cost.
Chimney	$1,200 to $2,500
Boiler plant	9,000 to 11,000
Four to six elevator plant	28,000 to 40,000
Heating appliances and piping	14,000 to 17,000
Electric wiring and switchboard	9,000 to 15,000
Engines and generators	8,000 to 10,000
Sanitation and fixtures	12,000 to 15,000

The effect of an increase in the cost of these appliances, if the same be productive of economy and advantage in operation, is very much less in proportion than its importance to the owner. For instance, the difference in the above comparative costs would amount to $26,000, say 4½ per cent. on gross cost of building, which would represent fully 50 per cent. more convenience and

the best appliances for economy throughout. It has, however, been difficult at times to induce owners to spend on economical appliances a small fraction of the sum laid out on non-remunerative outside or interior adornment.

The difference between a compound and a triple-expansion pump may be, say, $1,500, and be begrudged, while the difference between marble trim and other substitutes in the hall will be $15,000 and be freely expended.

These are difficulties met by engineers everywhere, yet in no case do they appear so forcibly. It is satisfactory to relate that such instances have been rather due to the present faulty system of letting general contracts inclusive of machinery than to the owners' short-sightedness as regards the advantage offered.

Equipment.

The value of office buildings to the business community is based on the improvements they offer, which are chiefly of a mechanical character, and further examination of the requirements will make it evident that the importance of the mechanical equipment considerably outweighs the value of any architectural features.

The details of direct interest to mechanical engineering science in these great buildings may be grouped as follows:

The character and arrangement of mechanical plant.

Elevator service and the limitation of the height of buildings by the proportions of same.

Piping and connections.

The power plant, chimneys, coal supply, feed-heating, and waste heat.

Heating, ventilation, and refrigeration.

Fire protection and water supply.

Electric lighting, wiring, generators, and storage batteries. Telephone and electrical services.

It will be obvious on a moment's thought that each one of these subdivisions is of sufficient importance, if treated with any fullness, to exceed the limits of a single paper. It is the object of the author at present to refer briefly to some of the problems connected with the arrangement of the mechanical plant of such an office building, as determined by the usual conditions which are prescribed.

Location of Piers, Etc.

The author's remarks are not intended to comprehend the details of the framing or building construction of these tall buildings, except in so far as they relate to the mechanical equipment required to operate them, but a brief *résumé* is necessary of the features of New York practice in respect to pier and foundation construction which bear on the subject of the arrangement of the plant.

The lower portion of New York City is composed of rock and sand, the rock being very irregular in outline, but roughly composing a ridge or hump on the line of Broadway, often dropping off abruptly. The sand is the floor of the glacial river-beds east and west, and is firm and affords a good foundation. At the extreme end of the promontory and along the west shore there is a good deal of made ground, but only one of the tall buildings with which the author has been associated trenched upon this, and in that case was carried below its level. Where the sand overlays the rock in a shelving form, the foundations are carried to the rock, but a number of buildings exceeding 200 feet in height have had their columns set on isolated concrete piers laid on the sandbed.

Such piers are usually 6 feet square at the base, are 2 to 4 feet thick; on top is set the shoe of the column upon a plate about 4 feet square.

The columns forming the walls are similarly treated, but the piers are usually united by a footing of concrete on which the curtain walls are run up. These piers have to be avoided in arranging foundations for machinery or in laying subsoil piping.

In one instance, that of the German-American Building, where the foundation was on sand, the Chicago method was followed, of spreading the bases over the entire area by a gridiron of I-beams, the lower series extending to a width of 11 feet, making the location of machinery a very difficult problem. The accompanying plan illustrates the arrangement of machinery effecting this result.

In the Bowling Green Building many of the columns could have been set on the bare rock, on the northern half of the building, but the concrete blocks were required by the regulation of the Department of Buildings, and the rock was blasted out for the

purpose. The rock shelved off towards the Hudson, and degenerated into a conglomerate, which proved to be watery, reaching to the original shore of the river within historic times, where made ground began. The natural water level is above the sub-basement.

In the Hudson Building, just across Broadway, the same depth of excavation did not expose the rock, and piles were driven about 20 to 30 feet through sandy clay to bedrock, the water level standing about 2 feet below the sub-basement level.

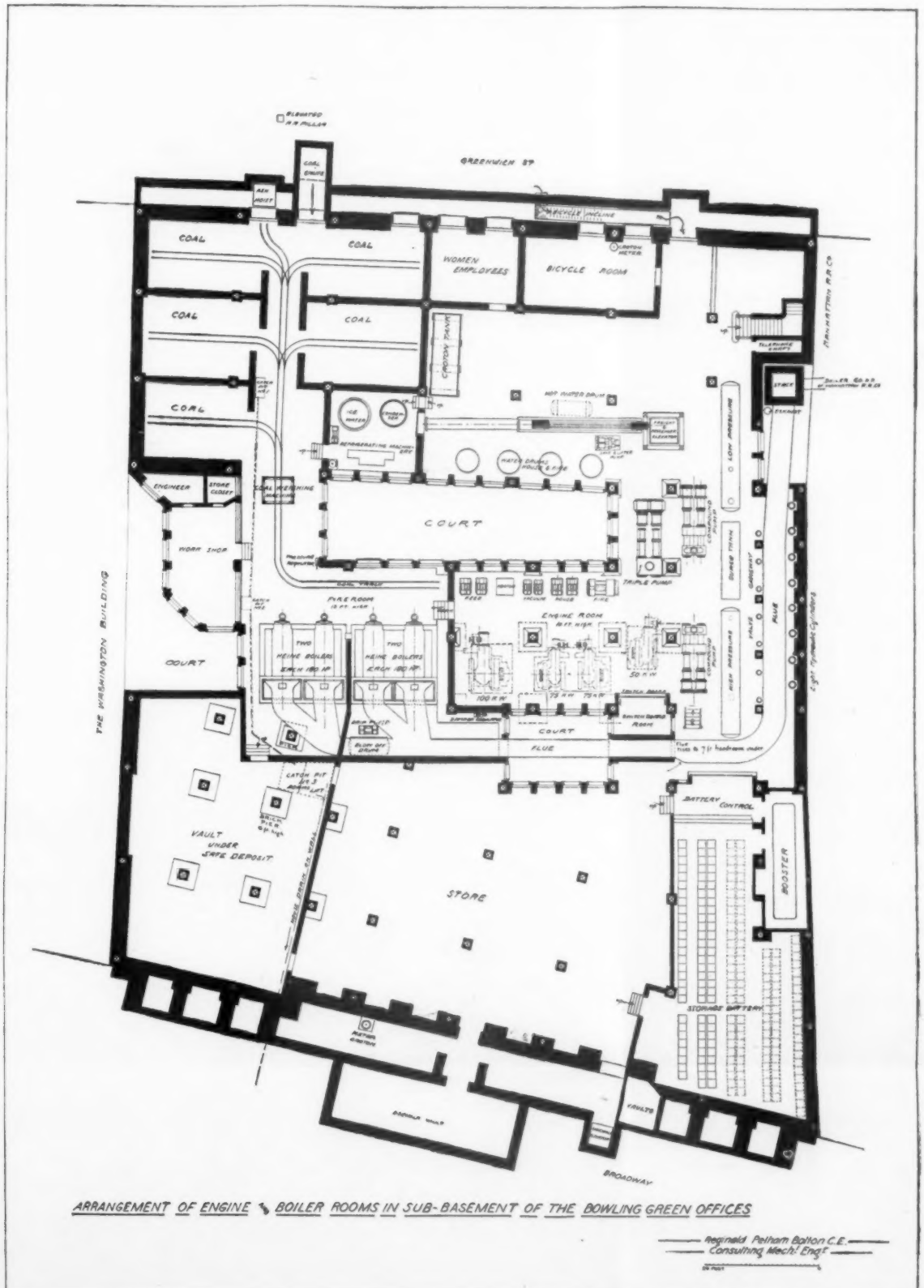
The method of support of this building, which was designed by Mr. Henry Hodge, M. Am. Society of Civil Engineers, consisted of a concrete bed formed on the pile tops, standing on which are exposed plate girders, 5 feet deep, across the sub-basement, on which the columns stand. Within the spaces thus formed, the plant had to be arranged, which was accomplished as shown in the accompanying plan.

In the Lord's Court building there is no sub-basement, but advantage was taken of the difference in level between the street and the Lord's Court at the rear to afford air and light to engine and boiler rooms. This building is carried on brick piers under each column in the usual mannér. In all buildings the columns and bases are cased in with fireproof brick or terra cotta.

Space.

The general practice now provides a sub-basement for the purposes of the plant, providing much more space than could be afforded nearer the street. Yet such are the demands of business that even sub-basement space is valuable, and cannot altogether be devoted to the engineer. Such sub-basements are also very difficult places to provide with air and light, for which a very early demand must be made, if they are to be secured. The height generally is not less than 10 feet, which, however, is insufficient for water-tube boilers; therefore the fireroom is usually excavated to a further depth, exposing the bases of columns and frequently going below the water line.

In laying out the four arrangements illustrated in Figs. 305, 306, 307 and 308, the guiding consideration has been to so locate the boilers that the fireroom will have a natural supply of light and air, and shall be readily accessible from the engine room. The author has departed in each instance from the practice of placing the boilers



ARRANGEMENT OF ENGINE & BOILER ROOMS IN SUB-BASEMENT OF THE BOWLING GREEN OFFICES

Reginald Pelham Bolton C.E.
Consulting Mech! Engr

FIG. 305.

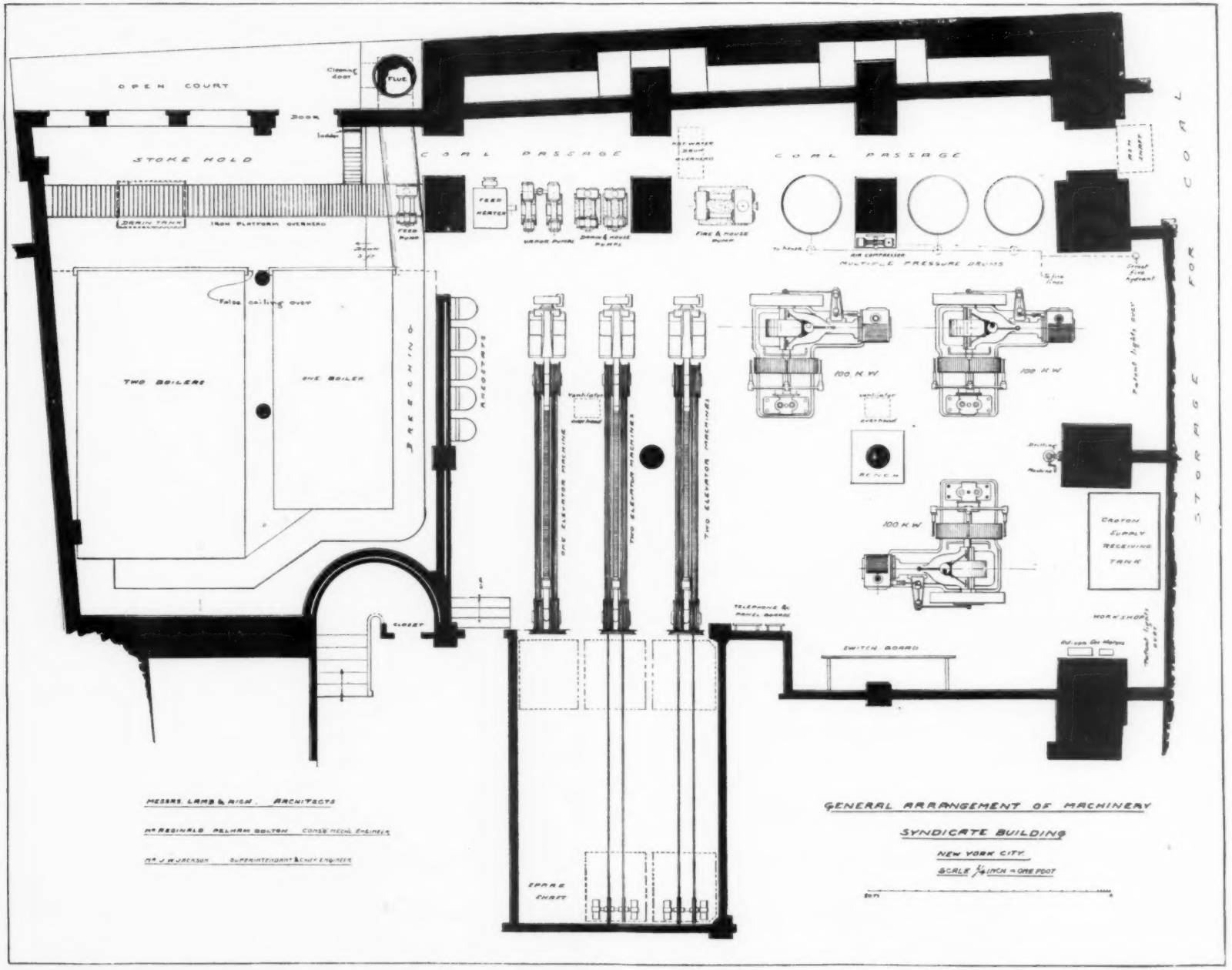

OPEN COURT
STOKE HOLD
COAL PASSAGE
COAL PASSAGE
COAL STORAGE FOR
TWO BOILERS
ONE BOILER
FEED HEATER
FEED PUMP
DRAIN TANK
IRON PLATFORM OVERHEAD
VAPOR PUMP
DRAIN & HOUSE PUMPS
FIRE & HOUSE PUMP
AIR COMPRESSOR
MULTIPLE PRESSURE DRUMS
ONE ELEVATOR MACHINE
TWO ELEVATOR MACHINES
TWO ELEVATOR MACHINES
100 K.W.
100 K.W.
100 K.W.
BENCH
CROTON SUPPLY RECEIVING TANK
SWITCH BOARD
WORK SHOP
CLOSET
SPARE SHAFT
MESSRS. LAMB & RICH. ARCHITECTS
Mr. REGINALD PELHAM BOLTON CONSULTING ENGINEER
Mr. J. W. JACKSON SUPERINTENDENT & CHIEF ENGINEER
GENERAL ARRANGEMENT OF MACHINERY
SYNDICATE BUILDING
NEW YORK CITY.
SCALE 1/8 INCH = ONE FOOT

FIG. 306.

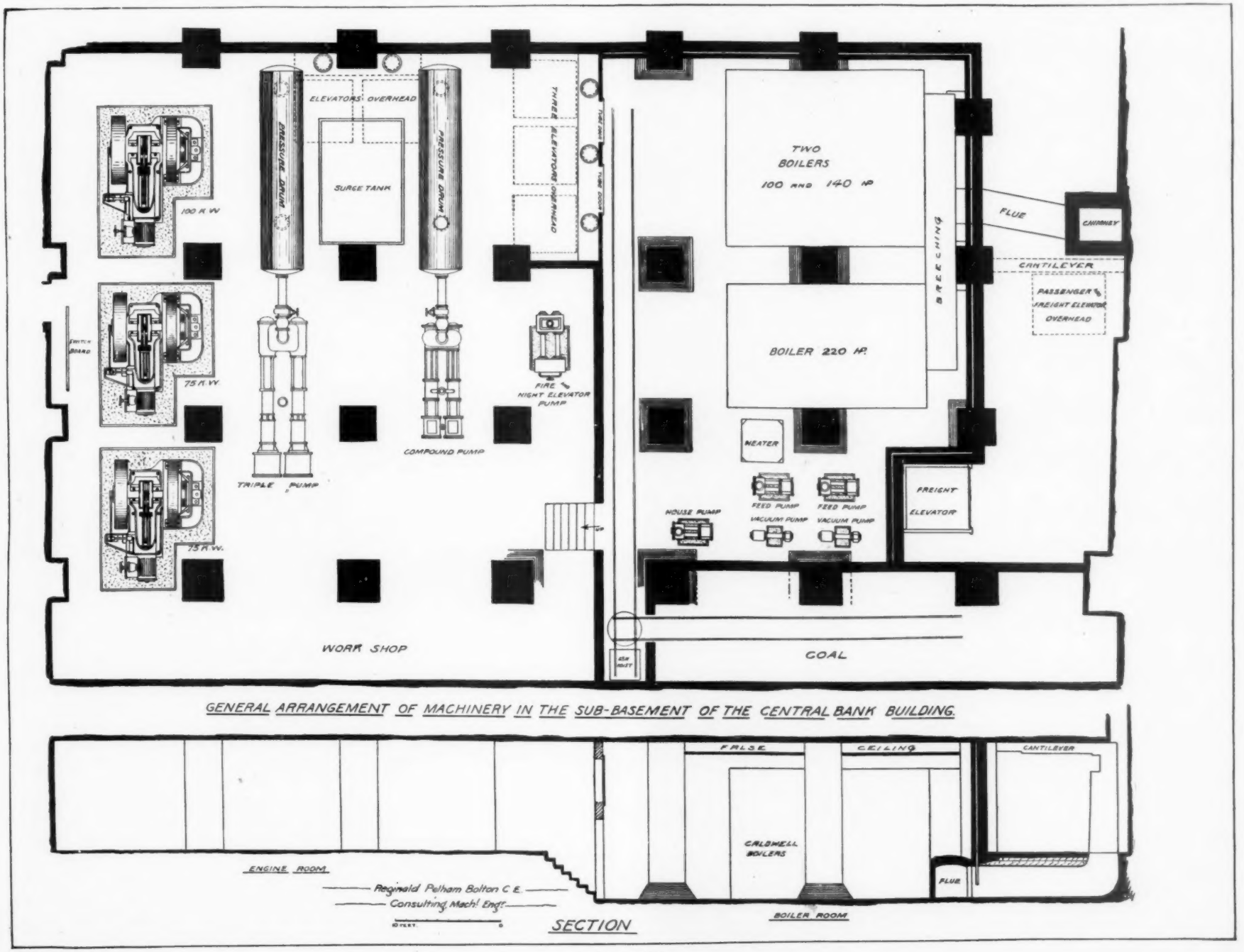

ELEVATORS OVERHEAD
PRESSURE DRUM
SURGE TANK
PRESSURE DRUM
THREE ELEVATORS OVERHEAD
TWO BOILERS 100 AND 140 HP
BREECHING
FLUE
CHIMNEY
CANTILEVER
PASSENGER & FREIGHT ELEVATOR OVERHEAD
100 K.W.
75 K.W.
75 K.W.
SWITCH BOARD
BOILER 220 HP.
FIRE & NIGHT ELEVATOR PUMP
COMPOUND PUMP
TRIPLE PUMP
HEATER
FEED PUMP
VACUUM PUMP
FEED PUMP
VACUUM PUMP
HOUSE PUMP
FREIGHT ELEVATOR
WORK SHOP
COAL
GENERAL ARRANGEMENT OF MACHINERY IN THE SUB-BASEMENT OF THE CENTRAL BANK BUILDING.
FALSE CEILING
CANTILEVER
CALDWELL BOILERS
FLUE
ENGINE ROOM
Reginald Pelham Bolton C.E.
Consulting Mach! Engr.
SECTION
BOILER ROOM

FIG. 307.

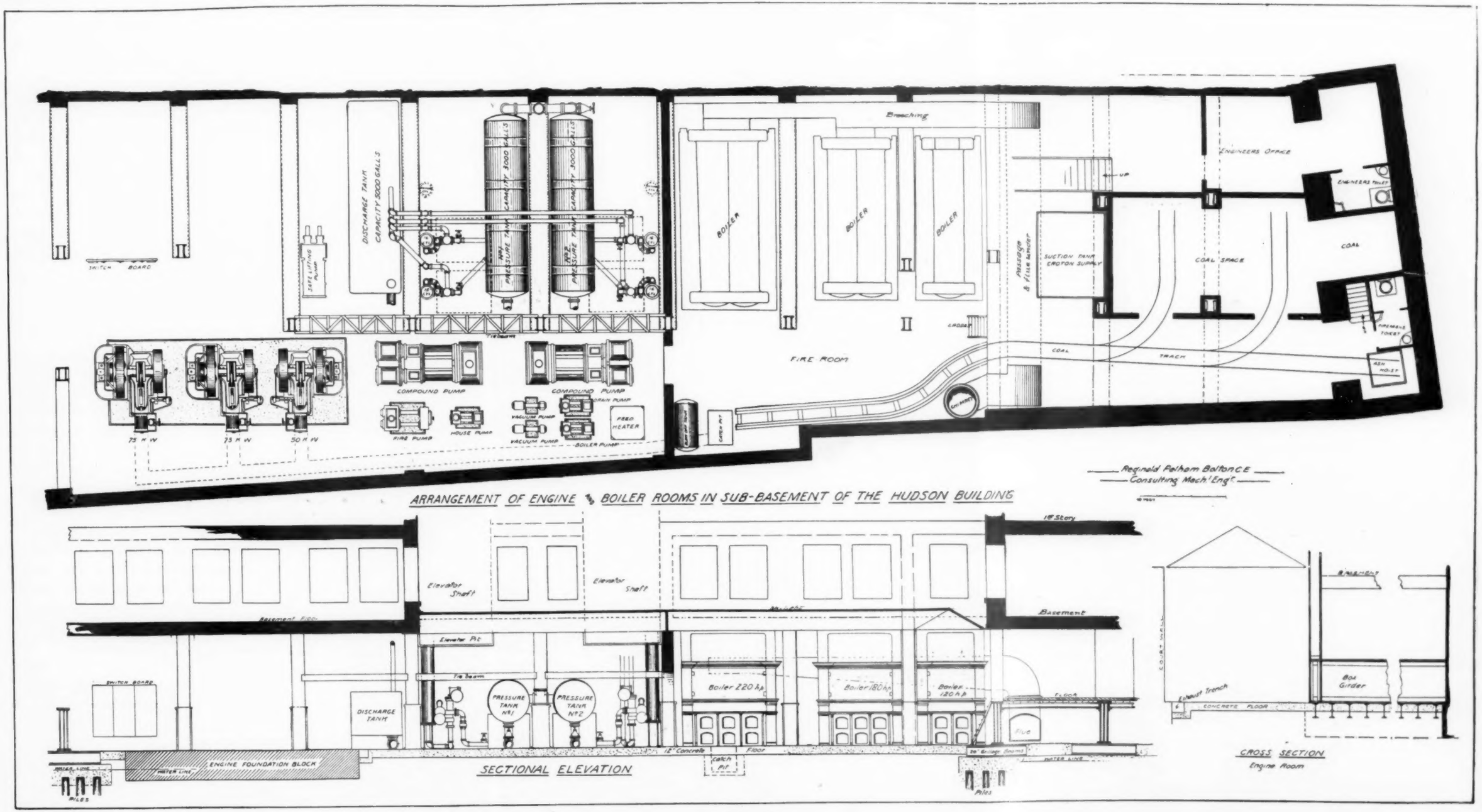

SWITCH BOARD
DISCHARGE TANK CAPACITY 3000 GALLS
SAFE LIFTING PUMP
No1 PRESSURE TANK CAPACITY 3000 GALLS
No2 PRESSURE TANK CAPACITY 3000 GALLS
75 H.P. IV
75 H.P. W
50 H.P. IV
COMPOUND PUMP
COMPOUND PUMP
FIRE PUMP
HOUSE PUMP
VACUUM PUMP
VACUUM PUMP
BOILER PUMP
FEED HEATER
Breaching
BOILER
BOILER
BOILER
FIRE ROOM
Passage & Flue above
SUCTION TANK EROTON SUPPLY
COAL SPACE
COAL
ENGINEERS OFFICE
ENGINEERS TOILET
UP
COAL
TRACK
PIPEMANS TOILET
ASH HOIST
Reginald Pelham Boltonce
Consulting Mech' Engr
ARRANGEMENT OF ENGINE & BOILER ROOMS IN SUB-BASEMENT OF THE HUDSON BUILDING
Elevator Shaft
Elevator Shaft
Basement Floor
Elevator Pit
Tie beam
SWITCH BOARD
DISCHARGE TANK
PRESSURE TANK No1
PRESSURE TANK No2
Boiler 220 h.p.
Boiler 180 h.p.
Boiler 120 h.p.
18" Story
Basement
FLOOR
Flue
12" Concrete
Catch Pit
Floor
20" Girder Beams
WATER LINE
ENGINE FOUNDATION BLOCK
WATER LINE
PILES
Piles
SECTIONAL ELEVATION
Exhaust Trench
CONCRETE FLOOR
Box Girder
CROSS SECTION
Engine Room

FIG. 308.

under the sidewalk, and has preferred even so long a flue as that in the Bowling Green Building, which is 240 feet in length, to a less advantageous position for the fireroom. It must be recorded that the location of the chimney is usually fixed beforehand by the architect. In the Bowling Green Building it could hardly have been in a worse position. In the Lord's Court building the shape is faulty.

The disposition of the main engines and pumps is chiefly decided by the position of the foundations of columns, and the necessity of access and repair. The arrangement of auxiliary pumps is such that they shall be so readily reached by the fireman that it shall be unnecessary for the engineer to stand by them at all times.

This also enables the engineer, at certain hours of light duty, to attend to the firing.

The relative position of the coal storage to the fireroom is of less consequence than its proportions. The fireman may easily attend to his own coal-passing if provided with a track. But unless the coal storage is large a coal trimmer must be kept to receive the daily supply. Trimming in the store costs, in certain buildings, 5 to 15 cents per ton. Such a large storage as the Bowling Green, which can contain 420 tons, has been found of advantage in reducing the price of fuel about 10 cents per ton, as it is not required to be delivered at any particular hour.

Buildings on Broadway are not permitted to receive coal or remove ashes and paper during certain hours, and consequently have to pay more for each convenience. The cost of removal of ashes under such conditions is 65 cents per truckload; but where the ash cans can be taken at any time, as on a rear street, the city removes without cost.

Labor.

A considerable variety and extent of labor is required in office buildings, and may be reduced by carefully planned arrangements and appliances.

The general superintendence is commonly delegated to a renting agent, who is paid by a percentage of 3 to 5 per cent. on the rentals received. The actual duty of superintendence then falls on an employee who has, generally, a number of such buildings in charge, which are therefore largely left to the mercy of

the janitors. A better method is followed in some buildings by the addition of the duty to that of the chief engineer, when much closer supervision over details of operation is possible.

The rates of wages per week are as follows in an average sixteen-story building:

One janitor, without residence	$15
One doorman, or hall porter	$10 to 12
One car starter, with uniform	12 to 14
Four to six elevator boys, with uniform	10 to 12
Three window cleaners	8 to 10
One toilet cleaner	8 to 10
Twenty to thirty scrubwomen (5 to 9 P.M., 6 to 9 A.M.)	6 to 8
One watchman	10

and the following in the engineering department:

One chief engineer	$25 to $31 50
One assistant engineer	20 to 22
One electric-elevator man	20
One electrician	18 to 20
One night engineer	15
One leading fireman	16
One assistant, or night fireman	13 to 14
One coal passer and ashman	12
One oiler	12

City Ordinances.

The chief engineer must carry a police certificate, and the assistant engineers generally do so, while some work on a fireman's certificate. Firemen's certificates are issued by the Police Department after an oral examination by the Chief Inspector of Boilers.

This department has power to inspect and test boilers operating over 10 pounds pressure per square inch, and to issue firemen's licenses for same.

Inspection is rarely exercised, and the test is simply the hydrostatic application of double working pressure. The inspectors have no authority to deal with improper settings, proportions, or connections, but do usually require, where more boilers than one are set in battery, two separate stop valves to each boiler, with a drip between the two, without which they decline to make the test.

The Department of Buildings has power to prevent the operation of any electrical power plant which does not comply with

its electrical regulations, for which it issues an operating license to the owner.

The departments of Health and of Buildings have excellent regulations regarding sanitary appliances, but the Health ordinance respecting the exclusion of steam and vapor from the sewers is constantly violated.

It is remarkable that the only trade employed in these large buildings which is subject to no regulation or supervision by authority is that of steamfitting. With the sole exception of a building requirement as to the protection of pipes concealed in proximity to woodwork, no demand can be made on them by authority, notwithstanding the numerous accidents which have been caused by faulty steam-piping.

Another matter demanding extension of authority is the discharge of heated air on sidewalks from the engine and boiler rooms of office buildings.

Street Supplies.

There are in New York extensive purveyors of electricity, both direct and alternating currents, and steam under pressure.

The latter extends over the greater portion of the downtown district. All, as well as gas and water, naturally demand consideration as to their use in operating plants.

The author, in comparing cost of these supplies with the expected results of an independent plant, has found, at existing prices, the advantage to be in favor of the latter.

Steam.

The steam supply is provided over the downtown portion of the city at a pressure of 70 pounds per square inch, while an economical use can be made in these buildings of 100 or 120 pounds pressure. The price of the steam supply is proportioned on the extent of its use, yet on largest and lowest basis it is in excess of the cost of local generation of steam. The case may be the reverse in smaller buildings, chiefly by reason of an economy in labor, but in the large office buildings a staff of men is necessary for maintenance and emergency work, and the wages of one fireman only represents about 16 per cent. of the full bill in an average building.

The New York Steam Company have abandoned their attempt to regain the condensed water from these buildings, their efforts having proved futile with their past apparatus. Consequently their steam is raised under less favorable conditions than is the case in the large buildings.

It would seem that successful extension of this steam supply in future largely depends on the adoption of the modern methods by which this desirable feature can be assured, and their prices of steam thereby reduced. The difficulty of disposal of the hot condensed water from the large buildings is an objection to the use of this service.

Gas.

At the present price of $1.10 per thousand cubic feet of gas a brake horse-power-hour can be obtained by a gas engine for a cost of 2 cents. This is certainly a low figure, and one that enters very closely into competition with steam. There are, however, certain disadvantages in the use of gas engines for these large buildings, which have hitherto deterred the author from their adoption. The great size of the units required is one, the liability to noise and to vibration is another. As the machines would be situated in the basement, the least vaporous smell of gas would be promptly objected to by tenants.

The question of irregularity of motion in lighting work can, no doubt, be disposed of by combination with a storage battery.

The same remarks apply to the use of the petroleum engine, which, however, is at present limited to units of not exceeding 50 effective horse-power. With these economical machines one brake horse-power can be obtained, at present prices of oil, for 1 cent per hour.

Electricity.

The electric supply of the illuminating companies forms the most serious competitor with an independent plant. At present prices for power the nominal rate is 10 cents per horse-power-hour by meter, and users of large quantities can scale this rate down to as low as 4 cents per electrical horse-power-hour on a use of 10,000 horse-power-hours per annum, with a still further reduction of about one-half cent if a storage battery be adopted.

But although the offer is of a temptingly simple and apparently advantageous character, a still further reduction in rates is re-

quired before the supply can compete with the results of a really economical independent plant. This reduction is probable, and, in combination with the storage battery and electric pumping for hydraulic elevators, the question will have to be reconsidered in the future. The charges must, of course, be increased by the cost of maintenance and interest on cost of the battery.

At the time of the establishment of the Bowling Green Building a careful computation showed, at the net price for lighting current of 8 cents per kilowatt-hour, including lamp renewals, and at the then price for elevator power of 7 cents per kilowatt-hour, an advantage in favor of an independent plant exceeding $5,000 per annum. The results of operation have shown a still larger advantage, as the plant was debited with the services of an engine-room staff of nine men, whereas it is now operated by only five hands.

The item which largely operates in this comparison is the utilization of exhaust steam in house-heating for the 100 or 120 days of cold weather.

The cost of a horse-power-hour for power, oil, and repairs, generated by an independent plant, may be, with economic arrangements, as low as $2\frac{4}{10}$ cents, and a fair figure, inclusive of interest and depreciation, is $2\frac{8}{10}$ cents; adding one-half share of all engineering labor, which is more than its due, a covering figure is $3\frac{8}{10}$ cents.

The installation of an independent plant has, so far, been decided by these considerations and by its relative superior economy. In future it is quite possible that this economy will be on the other side, but the use of such plants will, nevertheless, probably continue on the ground of the desirability of independence of outside supplies.

Such an argument recently received forcible illustration when the entire steam supply of the New York Steam Company was cut off by the action of a Water Department foreman, in turning off a valve in the main, the steam heat, elevators, and lights of upwards of four hundred buildings being cut off from 3 P. M. to 9 P.M., causing most serious disorganization and loss.

The above considerations are those which lead up to and affect the decision to employ and install an independent plant in office buildings, such as those described in this paper.

DISCUSSION.

Mr. William H. Bryan.—Mr. Bolton has "filled a long-felt want" in placing on record his figures as to value of ground space, dimensions of building, cost of mechanical plant, salaries of employees, cost of outside steam and electrical service, and the cost of generating power in isolated plants. Experience in large buildings in St. Louis is similar. In nearly every instance it has been found that electrical service can be generated more cheaply in an isolated plant than it can be purchased from an outside company, even at the special rates usually given to this class of service. The only exceptions are where there may be an abnormally high demand for lighting or power at a time when the plant of the central station is underloaded, under which conditions very attractive rates are made. The advantage of the isolated plant lies in the fact that for nearly one-half of the year coal must be burned for heating anyway, which heating can be done quite as satisfactorily with exhaust steam from power engines, with practically no additional cost for fuel.

The basement plans which Mr. Bolton submits are interesting. I was particularly impressed with the large coal storage space which is always provided, and the distance which the coal must be transported to reach the boilers. The use of so much valuable space for coal storage, and the necessity of repeated handling of the fuel, seem undesirable features. My own practice is to arrange the coal storage space immediately in front of the boilers, and make it large enough to hold only two or three days' supply. The coal is dumped directly from the street into this space, and fired directly into the boilers. This location usually affords ample light and air.

The great length of smoke flue, and the number of abrupt bends, would ordinarily be considered objectionable, but with a chimney 200 feet high or more the draft is no doubt ample to overcome the increased friction.

The advantages of the electric elevator in simplicity and arrangement of generating plant are well shown by comparing Fig. 306 with the others.

I would like to ask Mr. Bolton whether the storage battery belongs to the building or to the Central Edison Company. I had an impression that the battery in that building was a part of the Edison Company's system.

Mr. E. P. Roberts.—Not having had any personal experience in designing engineering equipment for buildings in New York City, I can merely note certain points in the article from the standpoint of one who has had experience in designing similar equipments in other cities.

I note that Mr. Bolton considers that the necessity often forced upon the engineer in specifying machinery which is not the most economical, all things considered, for the given conditions is due to the letting of general contracts inclusive of machinery, rather than the owners' short-sightedness as regards the possible, or rather, sure, advantage of machinery of greater first cost. If such is the case in New York, the owners have been educated up to a higher plane in such matters than exists in some other places. Very often the owner leaves all matters to the architect, who desires to erect a building having as handsome architectural features as possible, and still at the minimum cost. This is certainly proper, but in order to keep down the cost he prepares preliminary estimates for machinery possibly of a reliable character, but not unlikely less efficient than the case justifies. Very often he obtains his figures from contractors who desire to stand well with him, and who do not wish to be considered "high" men, and therefore give figures for cheap installations. If an engineer is afterward employed and advises more expensive material, it is much more difficult to obtain the necessary money than it would have been in the first instance.

If the client and also the architect not only appreciate the fact that the expenses of operating the machinery are a very considerable portion of the total operating expenses of the building, but also that a very slight modification makes a difference of at least 10 per cent. in such expenses, there would be greater probability that the most desirable plant would be installed, and that an engineer would be engaged at the start.

Men about to erect a building often think that the architect should know all about it, and that they do not need an engineer, much less one who has paid special attention to such matters. The architect himself does not usually appreciate the fact that an engineer is needed, and if he does, does not disabuse the owner's ideas upon the subject. The writer has had experience with both clients and architects who did appreciate it, but believes that they are rare. For example, we have had bids brought to us to advise which was the best to accept, and the machinery bid on

was so varied in size and character that no comparison could be made, and we have so stated. Nevertheless, the architect made a contract with one of the bidders, and if he obtained the best results for the least money it was a happy coincidence.

We have been asked to state what sizes of boilers were needed for a certain building, without giving us any information more than that a certain number of elevators were to be operated, and a fan for heating and ventilating, and also that there would be direct radiation. A request for information as to size of pumps, speed of elevators, and number of trips per hour, and size of fan and heating surface to be used, and the amount of direct radiation, and also request as to size of stack provided by the architect, and character of furnace, was met by the answer that it was considered that such refinement was unnecessary; all that was wanted was a "practical opinion." We believe we could have guessed, from having a slight knowledge of the building, within 50 per cent., possibly closer, but a comparatively small expenditure of time on our part, and of money on the part of the inquirer, would have eliminated the guessing and brought the data to within the limits of our ability to decide accurately upon the factors to be used.

I note Mr. Bolton's statement relative to begrudging money for compound or triple expansion pumps, and have had the same experience. I also note that for fan systems a common specification by architects is to state merely the number of times the air must be changed per hour in the building, or possibly in certain rooms, without any reference to the dimensions of the fan. Naturally, the smaller the fan the lower the bidder can make his price. We recently tested a fan taking 15 horse-power, and doing the work in accordance with the contract, whereas half the power would have been amply sufficient with a fan of proper size.

Also, as stated by Mr. Bolton, the location of the stack and its dimensions are very liable to be bad, sometimes the location necessarily so, though in other cases it might have been modified if the boilers and other machinery had been located conjointly by the engineer and architect. The matter of getting over the boilers is often a serious one, and if tubular boilers are used it is not infrequent to find that when new ones are desired they must be built in place.

Relative to the advisability of installing isolated plants, I believe that in nearly every case where there is a sufficient amount of machinery to necessitate the constant attention of a good

engineer it will pay to install electric light machinery. This is especially true where heating is needed for from six to seven months in the year. Where heating is needed for a shorter time the additional expenditure may or may not be justified. I have in mind, however, one plant where I advised, after careful consideration, and I believe properly, the installation of a gas engine and storage battery, which would be used during the summer months, and a steam engine to be used during the winter months, utilizing the storage battery at the same time. In case of break down of the steam engine and its dynamo the gas engine could be used. In case of break down of the gas engine, steam could be used if necessary. It was, however, very desirable that there should be no heat in the building during the summer months. It was also the case that a few times during the winter there would be more load than could be carried by either engine plus the battery, and at such times the gas engine could be used as an auxiliary. There was no question on anybody's part that this was the most desirable arrangement for the given conditions, but unfortunately it was found that the natural gas, which we were at first informed could be depended upon, would, not unlikely, give out within a comparatively short time, and that therefore a steam plant only would be installed.

I believe there will be a considerable growth in the next few years in the use of gas engines and storage batteries, and that a portion of such growth will be in isolated plants.

As an engineer interested in such matters, I thank Mr. Bolton for giving figures in connection with his paper, and when his paper was received I intended to write more fully on the subject, and also to present a number of figures, but time has not allowed.

Mr. Geo. I. Rockwood.—What is the present status of the great central station, considered as a competitor of the power plant located within a given building? If Mr. Bolton will treat on this question in detail I shall be under obligations to him. I should also like to ask Mr. Bolton if he has had any experience with storage batteries in office buildings, and if so, whether he advises their use?

Prof. R. H. Thurston.—This paper is an extremely interesting and instructive one and a model of its kind. I am particularly interested in the statements made regarding "city ordinances" and boiler inspection.

It is obvious to the most uninformed and inexpert that one of

these tall buildings is peculiarly subject to danger of destruction in case of explosion, and that the most rigid of precautions are here justified in regulations prescribing the character, the construction, the operation and the inspection and licensing of steam boilers and their attendants. If, as here stated, and as is no doubt the fact, "inspection is rarely exercised, and the test is simply the hydrostatic application of double the working pressure," the revision of the inspection law is as evidently imperatively demanded, and if the inspectors "have no authority to deal with improper settings" and other accessories, it is equally imperative that additional authority should be secured and exercised. While not at all probable, I imagine, it is perfectly conceivable that a steam-boiler explosion under one of these structures might wreck the whole pile, and it would always be liable to start a fire where combustible materials happen to be at hand. The heavy walls and foundations and the fireproof character of such buildings constitute a very effective insurance against such consequences; yet "it is the unexpected which happens," and one would think that the rules for regulation of the use of boilers in these basements and sub-cellars and under sidewalks would be the most stringent anywhere adopted. It would seem at least possible that the authorities might be instructed in such matters—of which they are possibly not well informed—and no better missionary work could be done than that of assisting these departments in perfecting their regulations and in securing ample authority to enforce them.

The cost of the unit space in these great piles, even in their basements, is so great that it may perhaps operate to insure the choice of the most compact forms of boiler, and those, fortunately, are the forms of greatest safety against disastrous explosion; but it is, one would think, late enough in the century to expect intelligent legislation on such matters and expert direction of the work of installation and of later inspection; insuring the adoption of forms of maximum safety and methods of effective inspection and regulation. There is no excuse, at this date, for mediæval construction or methods.

It would be interesting to know—and I have no doubt that the writer of the paper can inform us—just what difficulties stand in the way of such regulation and practice as expert practitioners, such as are many members of this Society, would approve. It has always been my conviction that this Society, officially and through

private action on the part of its members, has many opportunities for such missionary work as here offers. A good building law, applicable to this class of buildings particularly, a good system of inspection, and, above all, an effective system for securing a good official staff and exact and expert enforcement of the law and the regulations of the police and building departments, would seem to be greatly needed in many of our large cities and probably no less in New York than elsewhere. It is the business men of the community, and especially the members of the technical professions, who must compel the construction of proper regulations and their enforcement.

Mr. Frank M. Ashley.—In looking through this paper, I notice that, among the things treated of, the author speaks of the protection from fire. To my mind this is a subject of great importance, and one which I have carefully studied, and I believe that it is one which should have more careful attention from engineers in the designing of these buildings than it has heretofore received. I find in going through many of the large buildings, especially the older buildings, that instead of having straightway valves many of them are equipped with globe valves and many of them with gate valves of a smaller capacity than the pipes that lead to them. Now, where a valve is inserted in a stream of water the velocity is cut down very greatly.

I was present at the fire in Naething's restaurant, in Cortlandt Street, a very short time ago, and the Taylor Building, which is very near it, opened the streams from its reservoirs on the top of the building, and considering that those streams were intended as a protection against fire they were ridiculously small, and practically played no part in preventing the spread of the flames.

Some time ago I was much interested in what is known as the ball nozzle. I have the honor to be the first one to apply that principle to the spraying of water, and the National Ball Nozzle Company took the invention up, but unfortunately, on account of disagreements between that company and the American Ball Nozzle Company, they got to a point where they did not manufacture further. In testing nozzles of different designs and of different makes the importance of keeping straightways for the water, without either rings or projections on the interior of the pipes or valves, was shown to be a very important matter. It was found that by putting a globe valve between the supply and discharge, the velocity was cut down more than one-half, and

in many other nozzles which were designed to throw a stream of water from the end of the hose, the stream was cut down a great percentage on account of the turn or change in the water, due to the formation of the interior of the nozzle.

The mains in many of these buildings appear to me too small, and the connections and fittings which are used in many cases are decidedly inadequate, and should receive the careful consideration of engineers.

Mr. Bolton.—A question was asked me with reference to the storage battery in the Bowling Green Building. That is a battery which is the property of the Edison Electric Illuminating Company, and is a sub-station of theirs. It was, however, installed specifically for the purpose of taking the lighting load of the Bowling Green Building, which involves 6,000 lights, and which, had it been carried from their existing system, would have necessitated new mains from the Duane Street station, costing, I believe, about $48,000. The money was probably better expended by the Illuminating Company in installing this battery and in making the arrangement with the Bowling Green Building which they did, whereby they rent this portion of the cellar on very long rental at a low rate, and supply the building at a specially low rate as well, so that everybody's back is scratched. The battery has been a very great success. It has been described before other societies, and its operation has been such that the Duane Street station has been relieved of its heaviest peak of load during a certain portion of the day, and, I am told, has been shut down for three hours entirely, owing to the relief afforded by this battery.

With reference to the general subject of the use of storage batteries, I would like to say that I am a strong believer in their coming use, and most especially have been an advocate of their use in connection with electric elevators, a subject which will probably be dealt with in more detail later on. But I have installed storage batteries in other buildings, and notably in the R. G. Dun Building which is referred to here, in which the battery is located on the roof of the building, thereby occupying no space which is valuable for rental, and doing away with all the difficulty of getting rid of the fumes. I may say that the difficulty with the fumes is not an inconsiderable one, because the tenants of these buildings are exceedingly touchy, and if they smell the least thing that is disagreeable they want to get out at once. In the Bowling Green Building the battery is

vented by a 36-inch fan, blowing into the angle of the boiler flue and forcing the air up the chimney.

The question of the location of the coal bunkers in these buildings is generally fixed by the facility for getting the coal in, either on a back street or by the limits of the headroom. Very often these buildings, while they are deep on one street, will meet a street behind which is at a lower grade. That was the case in the Hudson Building which is illustrated here. There we only had about 7 feet headroom. It was manifestly impossible to deliver the coal by gravity if the coal bunker was removed any distance from that place; while the boilers, it will appear to you on examination, could not possibly go any nearer to that back street than the space they occupied. Also in the Bowling Green Building the same problem cropped up. Much difficulty is dismissed by storing a very large amount of coal, as in this case, where there is storage capacity of 420 tons, which has proved of the utmost value to them in extreme weather. During the recent blizzard they were not in the least danger of shortage of coal, while many buildings had to pay extravagant prices for coal to keep them going. I have in mind a building sixteen stories high, in Cedar Street, in which, owing to the futile way in which the machinery was placed, they had exactly 10 tons storage for operating a building whose demand is 7 tons a day. The coal is delivered down this small bunker to boilers which are located alongside the elevator pumps, which is not a very good place for the latter, and the result has been that that building has been partially closed down at times for the want of coal; and finally the owners concluded to drop their independent plant altogether and take steam and electric current from the street, which places them at a considerable disadvantage as regards cost.

One gentleman here referred to the very wide subject of fire protection. No subject is of more importance in these buildings. In fact, the first conception of them, I think, is that they shall be fireproof structures, and I may fairly say that no one has given more attention to the details of fire protection in tall buildings than I have myself, as I have fought now for four years past for proper appliances to be installed. Referring to the interesting occasion mentioned of the fire in a restaurant near the Taylor Building, I can very readily tell why those very small streams of water were delivered onto the roof. Most of those older tall buildings are provided with tanks on the roof, the water being

pumped up to them, and not into the mains, by a small pump which can scarcely be called a fire pump. The jet which is delivered from the tank is of course a gravity jet only. Now, from a hose on any floor underneath that tank the only jet that you can get out is due to the head between the tank and that floor. It is manifest that pouring a little stream such as would be procurable on the sixteenth story of a building is futile. I installed in the Bowling Green Building, and in nearly all of the buildings referred to here, the pressure-drum system, in which drums are located in the cellar and are used as standing pipes for house supply. A fire pump discharges into those, and all the drums are cross-connected. They are two-thirds filled with water and one-third with air. A large pipe is carried out to the street main, with a Siamese union on it, through which the fire department can pump into these stand-pipes. The risers connecting the hose pipes on different floors are carried up from that stand-pipe system as straight as possible, and are not disconnected from the house system. That I consider to be a vital point. If you do disconnect them, then the engineer or the janitor will surely drain the fire pipes, which will be full of air at the critical moment when they ought to be full of water. Consequently, I do not design them to be disconnected. But if the fire pump is started up, and a high pressure is required, which may go up into hundreds of pounds per square inch, a check valve shuts down on the house connection and prevents that high pressure reaching the house fixtures. With such an apparatus I have thrown a 60-foot jet on top of a 200-foot roof, in the presence of the representatives of the fire department of New York City, which was described by them as a good jet in putting out a fire. This subject will bear considerably more detail, and I hope on some other occasion I may be able to devote more time to it.

I would like to add that if Mr. Roberts' remarks could be added to my paper I would very willingly adopt them as part of my own experience and my own wording. I have been through all this unhappy experience of having to put in inferior apparatus, while marble fronts and trimmings of all kinds are going into the halls, and caryatides and other animals are being fitted on the front of buildings. We have had cases where a building is at a permanent disadvantage of many tons of coal per annum, and having to pay extra wages for all time because the owner wanted to spend money on these monu-

ments of ingratitude which stand on the front porches of these buildings.

Mr. Rockwood.—May I be permitted to question Mr. Bolton a little further in regard to storage batteries? His answer dealt with their use in connection with electric elevators, but not with reference to lighting.

I had the idea—and have put it into practice in one case—that a storage battery might be a wise investment, notwithstanding its heavy depreciation account, if used with a lighting plant, the load being handled in this way: A battery large enough to carry, let us say, one-half of the lighting load during the period of greatest demand could be "played into" by an engine of smaller size than would otherwise be necessary, during the entire day. Thus the exhaust steam from that engine would be constantly available for heating the building. Then at the lighting time the battery can be run on its own circuit and the engine and its generating apparatus be applied to the balance of the lighting load. Whether the interest and cost of maintenance of the battery would nullify the economy of heat I don't yet know.

The need for storage batteries in these tall office buildings is a growing one, and should be amply discussed.

*Mr. Bolton.**—To Mr. Rockwood's questions I can only reply here that the subject of storage batteries is an important one, and of too wide a nature to deal with in a discussion. I have the intention of bringing up results obtained with the use of the battery in connection with elevators, but this is deferred until the plant shall have been in operation for a sufficient length of time to demonstrate its endurance.

An illustration of the difficulty presented in the cost of outside sources of supply is afforded by the fluctuation in cost of two of those sources of power mentioned in the paper, since its reading.

Gas has been temporarily reduced to 65 cents per thousand cubic feet, and oil has been advanced nearly 100 per cent.

Professor Thurston's remarks are extremely practical, and I am glad to have drawn his attention to the subject of the absence of proper regulation, by authority, of boilers and steam-fitting.

I have always maintained that the Department of Buildings

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* Author's closure, under the Rules.

should be armed with authority on these subjects, but in every attempt which has been made to amend or improve the present building laws no representation of mechanical engineers has been provided on the committees dealing with the subject, and until this is done it is hopeless to expect attention to be given to the matter.

DCCCXXII.*

THE CENTRAL HEATING PLANT OF THE UNIVERSITY OF WISCONSIN, AT MADISON, WIS.

BY STORM BULL, MADISON, WIS.

(Member of the Society.)

THE central heating plant described in the following paper is not a model one, and the reason why it has been thought worth the while to present this description of the plant, with tests of the same, to the members of the American Society of Mechanical Engineers must, therefore, be another one than that of setting it up as an example to be followed. The reason ought to be found in the fact that in laying out and enlarging the plant difficulties had to be met and solved, which it might be interesting to have described, even to those who have been in position to lay out a model plant. Furthermore, published tests of Western plants of this character are not numerous, and, as in steam engineering only a large number of tests serve the purpose of giving a guide for the future, the following paper may also in this direction contribute something to the experimental knowledge for which our profession hungers.

The principal difficulty met with in laying out the central heating plant of the University of Wisconsin was that a boiler house already existed, from which power and heat were furnished for three adjacent buildings. The location of this boiler house was hardly the best one even for these three buildings; but when it was determined by the authorities to heat all the buildings from a central plant, it was at once seen that the location was a bad one for several of the buildings. But so much money had already been expended on the boiler house, it being a very substantial and somewhat ornamental building, together with the chimney, it was determined by the regents of the university, in 1894, that the old boiler house should be enlarged and remodelled so

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

that all the buildings of the university—except those of the Agricultural Department—could be heated from it. The accompanying plan of the boiler house and buildings (Fig. 309) shows their relative positions; and it will also be seen, from the numbers placed next to the various buildings, that these are located on the slope

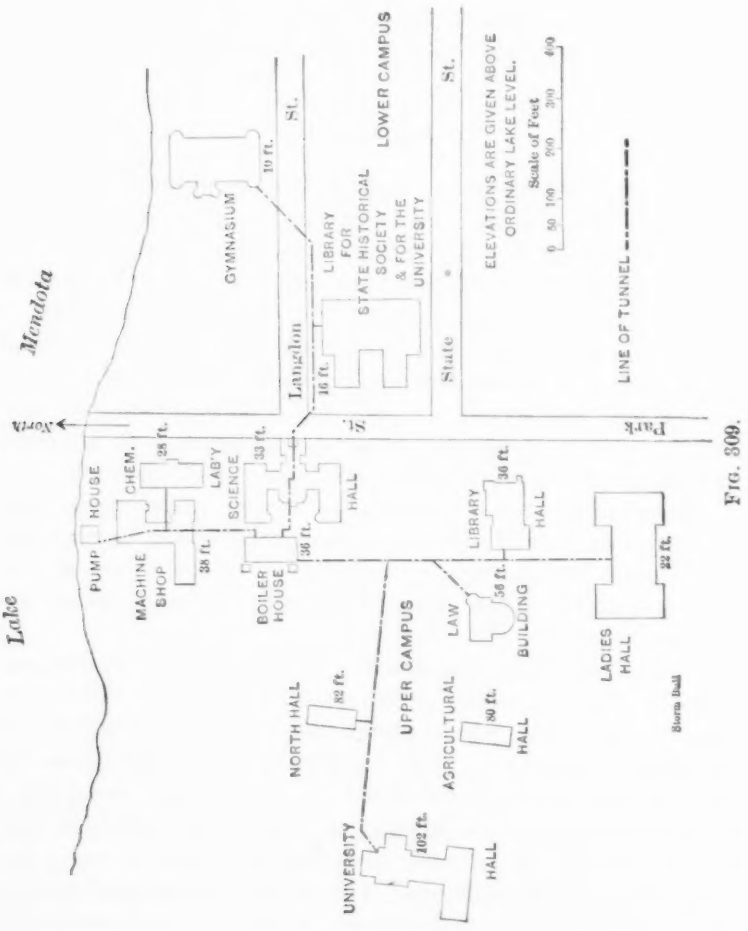


Fig. 309.

of a hill, the boiler house being about one-third way up, so that several of the buildings are on a lower level than it. The proper location for the boiler house would have been on the lake shore, both because all return water from the several buildings could have been returned by gravity, and also because of the hauling

of the coal and the disposal of the ashes. But, as stated above, the location of the central heating plant was not determined with simply the future economy in view, but because of previously invested capital, and things had to be arranged as best they could.

The original boiler house was a square building about 50 feet each way, and, in order that Science Hall, the chemical laboratory, and the machine shop might be heated from it, the floor had been sunk about 13 feet below the surface of the ground

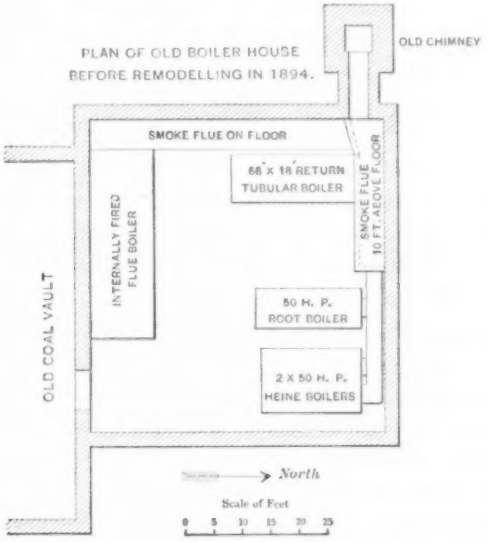


FIG. 310.

in front of the building. It contained, in 1894, when the writer of this paper was instructed to remodel the plant, the following boilers: One large internally fired flue boiler, 125 horse-power; one 66-inch by 18-foot return tubular boiler; two 50 horse-power Heine boilers; and one 50 horse-power Root boiler. These two Heine boilers furnished the steam for the engine in the machine shop and for the pumps at the lake shore. In addition to the buildings mentioned above as being heated from the boiler house, the Law Building had been connected with the boiler house by a tunnel in 1892; and it was at this time that the 66-inch by 18-foot boiler had been installed in the boiler house for the purpose of furnishing steam for heating this building, which had just been

completed. The boiler house was also connected with the pump house, machine shops, chemical laboratory, and Science Hall by tunnels, the location of which is shown on the general plan, and which had been built in 1885–86, when the boiler house and these buildings had been erected. From the accompanying sketch (Fig. 310) arrangement of the boilers in the boiler house at that time will also be seen; and it hardly needs to be stated that the arrangement was a very bad one, both as to convenience and utilization of space.

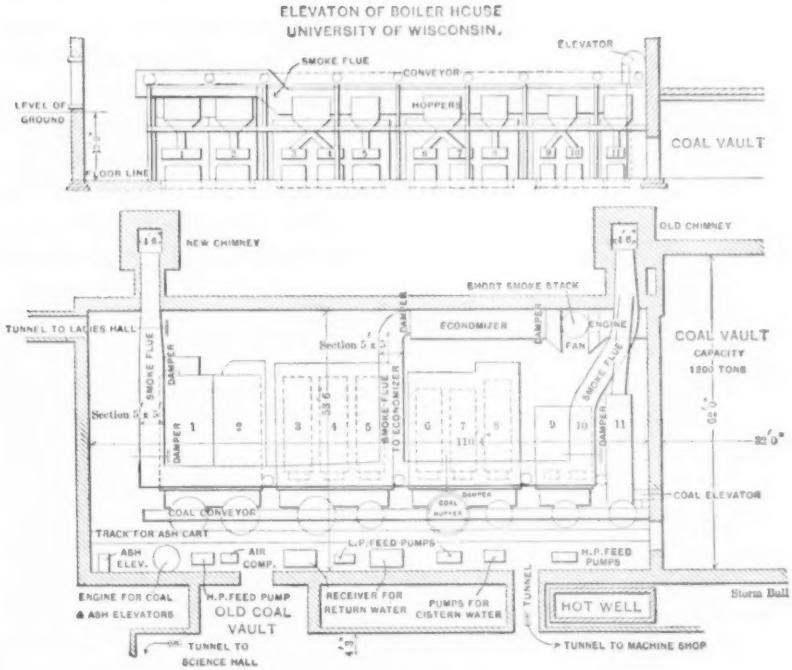
The gymnasium and armory of the university were nearing completion in 1894; but the plans of this building provided for two 60-inch by 16-foot return tubular boilers to be placed in the building, and the boilers had already been bought by the contractor, when the regents of the university determined to heat the building from the boiler house. Two of the old buildings of the university, North and University halls, had up to this time been heated by stoves and hot-air furnaces, and the regents the same year determined to put new heating apparatus into these two buildings, the steam to be furnished from the boiler house which now was named the central heating plant.

As a consequence of these changes the old boiler house had to be enlarged to such an extent that there would be room for boilers sufficient to furnish steam for three additional buildings, of which one especially was quite large, viz., the gymnasium, containing about 1,700,000 cubic feet of space. In addition it was determined to make the building so much larger that it might house several more boilers, so that other buildings might be heated from it in the future. In 1896 Ladies' Hall was connected with the central heating plant, and in 1898 the new library, both times an additional boiler being installed. At the present time there is still room for a boiler of 200 horse-power capacity, which space probably will be needed this year if, as seems probable, a new engineering building will be erected the coming year.

The arrangement of the central heating plant as it is at the present time is shown on Figs. 311 and 312. It will be noticed from these that the boiler house was a little more than doubled in size by the change in 1894, and that the boilers were arranged in a row, setting them for convenience in batteries of two or three. One wall of the old building was removed, the roof being carried on columns, and the coal vault was placed on the opposite side of the building to what it was before, this being done because of the

greater convenience of hauling the coal, which all must be hauled up between the chemical laboratory and Science Hall. For the same reason it was placed below ground, bringing the floor of the coal vault on the same level as the floor of the boiler house. The old chimney was a square one, the height being 138 feet from the boiler-house floor and 4 feet 6 inches inside. It was deemed

FIG. 311.



PLAN OF BOILER HOUSE
UNIVERSITY OF WISCONSIN.

FIG. 312.

sufficient for the enlarged plant, and no changes were made with it. But the smoke flues leading to it could not be used, both because of their size and location, and an entirely new breeching had to be arranged. One of the reasons for determining on the installation of a central heating plant was that it would be possible to use bituminous coal instead of anthracite, as was being used in the separate buildings, and without suffering from the

smoke and dirt to such an extent at least as would have been the case if bituminous coal had been used for heating the separate buildings. In the old boiler house anthracite had to be used for the old internally fired flue boiler, whereas bituminous coal was used for the other boilers, with the result, however, of creating quite a smoke nuisance, as there was no provision made for the "consumption" of the smoke. One of the conditions for the installation of the central heating plant was therefore that bituminous coal should be used, and that the plant should be so arranged that there should be very little smoke. A thorough investigation of the smoke question soon convinced the writer that several of the mechanical stokers, besides the Hawley down-draught and other arrangements, will, if properly taken care of, and if the boilers are not forced much, produce a fire which is reasonably free from smoke. In view of the saving of labor the writer determined on the use of a mechanical stoker, rather than, for instance, the Hawley down-draught furnace, and for the same reason, and because of the cleanliness, coal-conveying machinery with coal-storage tanks above the boilers was also made an essential part of the programme. As the floor of the boiler house was some 13 feet below the surface of the ground, an ash elevator was also a necessity; but the writer abstained from installing an ash conveyor, as the repairs on such a conveyor seem to neutralize more than the saving in labor realized by its use. A track with an ash dump-cart was determined upon in the place of the conveyor. In the old plant there had been no feed-water heater, but as will be seen from the plan a Green economizer of 400 horse-power capacity was installed, and in connection with it a fan with short smokestack for mechanical draught, this latter arrangement mostly for experimental purposes. It may be said here, in parentheses, that a short smokestack is out of place where there are buildings near by, as the gases escaping from the chimney, even if there be no smoke, are being delivered at such a low level that they cannot help being carried into the adjacent buildings, which, of course, is very objectionable.

Steam had to be furnished from the central plant both for heating and power purposes, and the question arose whether it would not be best only to carry steam of high enough pressure for power purposes at the central plant, and to reduce the pressure at the various buildings to use the steam for heating. On account of the fact that the boilers were considerably higher than the

return in several of the buildings to be heated, and in fact because all of the buildings were situated at such various levels and distances as compared with the boiler house, it was, of course, impossible to use a common reducing valve at the latter place for all the steam used for heating, and the only other alternative was to use a certain number of boilers for making steam for heating purposes of sufficient, but moderate pressure, so that it might be used for running the fan engines at the various buildings and otherwise be reduced in pressure at the buildings to serve for heating. This latter alternative was the one adopted, and for the following reasons:

Various members of the Board of Regents of the university objected to the use of "*high-pressure*" steam in the buildings in which only ordinary janitors, who are not engineers, had to take care of it; consequently a steam of less pressure than that used for the pumps at the lake shore or for the engines in the laboratories had to be used. At the same time the architects, who had planned the heating systems of several of the buildings, had provided engines for running the ventilating fans, which engines required steam of at least 30 pounds pressure, which pressure again would be too high for the direct radiation. If, therefore, a common reducing valve had been used at the central plant, it could not reduce the steam to less than 30 pounds pressure, necessitating an additional reducing valve in each of the buildings, or a separate pipe for taking the steam for the fan engines would have been necessary. Although the danger from using 30 or 40 pounds steam in the buildings is not very different from that of using 100 pounds steam, the ordinary pressure carried in the central plant for power purposes, yet to the authorities this difference seemed very important, and accordingly the following arrangement was adopted: All the boilers in the central plant were to be connected with two separate headers, one for high-pressure and the other for low-pressure steam, the first one to carry about 100 pounds and the latter about 40 pounds pressure. The high-pressure steam was to be carried to such buildings only in which steam was used for power purposes exclusive of fan engines, and the low-pressure steam to all buildings which were to be heated. It is perhaps necessary to add that the writer is of the opinion that it would have been poor economy to have carried in the tunnels leading to the various buildings steam of such low pressure that it could have been used for the direct radiation without being reduced in pressure,

because it would have necessitated such large pipes, which would have made the loss by radiation in the tunnels greater than it would be with smaller pipes carrying steam of higher pressure, although necessarily also of higher temperature. With the system adopted for this plant it is possible to keep the size of the pipes quite small, so small in fact that a considerable loss of pressure must be expected at the end of the main, which reduction of pressure, however, does not interfere with the working of the heating system of the building, as it is only necessary to carry a correspondingly higher pressure at the central plant, and because each building is independent of every other because of the reducing valve found in each building. At this place it might also be pointed out that experience in this plant has shown that it is preferable to have a separate reducing valve for the indirect tempering and heating coils, in order that a higher pressure may be carried on these than on the direct radiation, this to facilitate the circulation in these coils in very cold weather. The sizes of mains in the tunnels were necessarily calculated so as to carry sufficient steam to heat the buildings in the most severe weather. But especially in the fall and spring there is a considerable length of time in which but very little steam is used for heating, but always a certain amount for ventilating the schoolrooms, etc. At those times the large mains are altogether too large, and the percentage of loss of heat in the tunnels is excessive. In other buildings, like the gymnasium and armory, and also Ladies' Hall, warm water is a necessity the year around, and this warm water is obtained by steam from the central plant, and for this purpose also the original steam mains were too large. Because of these reasons separate small mains were run from the central plant to the various buildings and were connected at the several buildings with the large mains in such a way that the steam coming through the small pipes could be used for heating the buildings, running the fans or heating water as the special case required. The original cost of these "summer" mains was quite an item, but the saving in fuel during the past five years has more than paid for them. Another great advantage gained by having these two separate mains to each of the buildings is that in case the large main should absolutely need repairs, the smaller one can be used and will furnish sufficient steam to keep the building from freezing, at any rate.

Another important question affecting the economy of the plant

was that of the return-water from the various buildings. As will be seen from Fig. 310, there are several of the buildings which are lower than the central plant. After a thorough investigation it was conclusively proven that it would pay to pump the water back through the tunnels to the central plant, even if the distance was as great as from the armory, nearly 900 feet. The returns are all connected to closed receivers in the central plant, which receivers again are connected to the hot-well. Because of the two different pressures carried on the boilers, it was necessary to provide for two different feeding systems for each boiler, with a feed pump for each system. The low-pressure feed pump takes its suction from the return-water receivers directly, whereas the high-pressure pump has its suction connected with the hot-well. Into this latter all water required beyond the return-water is pumped, and from it the water is pumped through the Green economizer before it enters the high-pressure boilers. The additional water required is principally obtained from a number of rain-water cisterns located near the various buildings, the capacity of all of these together being about 6,000 barrels, most of the water running by gravity to the central plant. Quite frequently, however, water from the lake has to be used, but as this is comparatively soft water, and as it is nearly always heated up to from 250 to 300 degrees Fahr. by being passed through the economizer, very little scale can form even in the high-pressure boilers. As a result of the good quality of the feed-water used for the boilers very little or hardly any scale has ever been observed in the boilers, and no compounds are being used. It is impossible to estimate accurately what portion of the feed-water is return-water; but it is safe to say that more than three-fourths of it is of this origin during the winter months, as nearly all the exhaust steam from the various engines, etc., is used for heating purposes. During the summer months, when all the steam is used for power purposes, no return-water is obtained. In the central plant there has recently been installed a feed-water heater which utilizes the exhaust steam from the various stoker engines, feed pumps, the conveyor engine, and from the small engine which runs the dynamo which furnishes the light for the building. This feed-water heater discharges into the hot-well. As will be seen from the data of the tests, which are given later on, the temperature of the water of the hot-well reaches 170 degrees Fahr., which fact is due to the feed-water heater, and also to the discharge of a num-

ber of steam traps which take care of the condensation in the mains, etc., in the boiler house.

All of the return-water from the several buildings is discharged into two closed receivers in the central plant. There are not separate returns from all of the buildings to these receivers, several being connected into one at convenient places. However, it has been deemed best to have separate return pipes from those buildings from which the water has to be pumped back. Because of the fact that each building has a separate reducing valve and that therefore the steam pressure used in each building may be different from that used in every other, it was necessary to discharge all return-water through steam traps into the returns. It might have been possible to avoid these traps if open receivers had been used, but it certainly would have resulted in a great loss of steam from these receivers, and it is probable that the present return pipes would not have been large enough to relieve the various heating systems in several of the buildings from a too great back-pressure. As the returns now are arranged the heating apparatus of each building is entirely independent of that of every other building. This is in itself of great advantage, as the heating plants of the various buildings have been designed by several different parties, and, because of differences in the design, require steam of different pressures. This is especially true of the buildings in which there are ventilating fans, as it has been found necessary to use steam of quite a little higher pressure for the tempering coils than is used for the direct radiation under similar circumstances.

The central heating plant, as now equipped, contains the following boilers:

One Aultman & Taylor (Babcock & Wilcox type)....	200 horse-power.
One Standard boiler (water-tube).....	150 "
Two Heine boilers, 50-horse-power each.....	100 "
One Root boiler.....	50 "
Four 66-inch by 18-foot return tubular boilers.....	500 "
Two 60-inch by 16-foot " " "	160 "
Total, eleven boilers.....	1,160 horse-power.

Of these boilers, the Root boiler is used exclusively for experimental purposes, and is not connected with the steam mains in the central plant, but with an experimental engine in the steam laboratory in Science Hall. The boilers are all equipped with

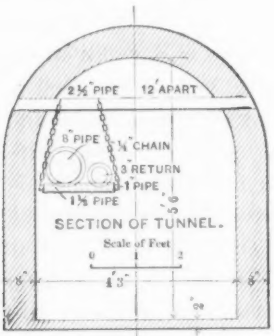
Roney mechanical stokers, for the working of which there are provided five small oscillating engines. They are all connected with both the high and low pressure mains and have two separate feed pipes, which, however, are connected together a foot or so from the boilers. There are two low-pressure feed pumps—one for reserve—and also two for the high-pressure feed, and each pump is provided with a hot-water meter. In addition to these pumps there is one small duplex pump for taking the water from some rain-water cisterns, which are lower than the central plant, to the hot-well, and there is a small automatic duplex air compressor, which furnishes the compressed air for the Johnson & Powers heat-regulating systems used in five of the buildings heated. The central plant contains, in addition, a 10-horse-power upright engine for driving the coal-conveying machinery, one 5-horse-power automatic engine driving a 60-light 16-candle-power dynamo, and the engine which drives the fan for the mechanical draught. It has already been mentioned that there is a Green economizer, which is of 400-horse-power capacity. The feed-water heater is supposed to take care of the exhaust steam from 200 horse-power.

When, in 1898, the 200-horse-power Aultman & Taylor boiler was added, it became necessary to increase the chimney capacity, as the old one built in 1885 was hardly sufficient for the boilers already there. In this connection it should be borne in mind that at no time are all the boilers in service; there is always at least one in reserve, so that the boilers may be washed out at frequent intervals or repaired if necessary. It would, of course, have been desirable to have built a new, larger chimney, which could have taken care of all the boilers. But because of the expense, it was necessary to keep the old one and only to add another one, which, to make things somewhat symmetrical, was made of the same height as the other. The new one was built with an entirely independent firebrick lining, whereas the old one was a solid unlined shaft. The breeching was rearranged, as shown in Figs. 310 and 311, in such a manner that either chimney may be used for any or all the boilers. Figs. 310 and 311 also show the location of the economizer. It will be seen that it can only be used in connection with the old chimney, a disadvantage which it seemed impossible to overcome.

The coal elevator takes the coal, dumped into the vault from the coal wagon on top of the coal vault, up to the conveyor,

which runs lengthwise of the central plant and carries the coal to the several hoppers placed in front and above the boilers, from which the coal slides through chutes to the stokers. The coal vault is used as a storage room, in case strikes or a snow blockade should prevent the regular delivery of coal. The ash elevator is placed in a corner of the building; the ash cart, into which the

FIG. 313.



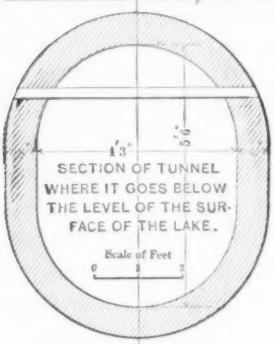


FIG. 314.

ashes have been shovelled, being dumped, so that the ashes need not be handled any further till they reach the ground above, from where they must be hauled by teams.

The tunnels through which the steam pipes and returns are carried to the various buildings have a total length of about 3,200 feet. Except those parts which connect the central plant with Science Hall, the machine shop, and the chemical laboratory, they have a section as shown on Figs. 313 and 314. This section is hardly large enough, at least for the portions near the central

plant, where the pipes are both larger and more numerous than in the other portions. If the choice of the size of the tunnel had been left to the writer it would have been made longer, but, unfortunately, the tunnel to the Law Building had already been built when he took charge, in 1894. The older portions of the tunnel, leading to Science Hall, etc., are larger in section, but not

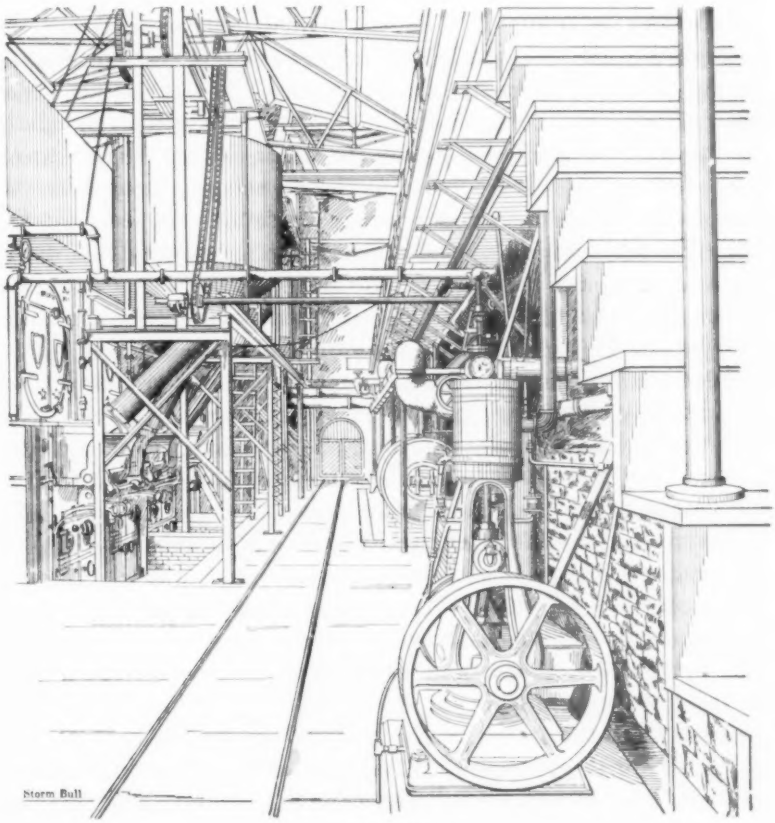


FIG. 315.

too large. They have, however, an elliptical section, which, because of the round bottom, makes it difficult both to walk through them and work in them. It should be stated that it has been found possible to make all kinds of repairs in the tunnels built as shown in Figs. 313 and 314; but the room is cramped, and it would be better if the section was about one foot larger each way. Fig. 314 shows the section of the tunnel where it goes below

the level of the lake. In order to make it tight at such places it was necessary to substitute a circular bottom for the flat one used elsewhere. At those places where two branches of the tunnel join, enlargements of the same have been made. This is necessary to give room for the numerous fittings, like valves and expansion joints, required at those points. And at a great number of places manholes have been placed, both for ventilation when work is being done in the tunnels and to facilitate the entrance to the tunnels. These manholes are very necessary during the summer, as they must be kept open almost continuously to prevent the condensation of the moisture of the air in the tunnels on the covering of the pipes, which condensation soon would ruin all the covering, notwithstanding any kind of painting of the same. The average depth of the crown of the tunnel below the surface of the ground is about 3 feet, varying between 7 and 2 feet.

The expansion of the steam and return pipes in the tunnel could only be provided for in a few cases by elbows and offsets, and a large number of the old-fashioned expansion joints are being used. By carefully anchoring the pipe in every direction at the place where the expansion joint is placed, and by making it possible for the pipe to expand freely in the other direction, experience has shown the writer that these much-decried expansion joints may be a source of very little annoyance. It has been made a rule to repack these joints once a year, not because it was certain that they needed a repacking, but simply as a precautionary measure, and the result has been that there has not been any leaky expansion joints. Sometimes these joints have been screwed up a little during the course of the year, but only because the engineer happened to be in the tunnel, at the place where there was an expansion joint, for the purpose of doing some other work. The reason for these good results the writer sees in the careful anchoring and alignment, and he is confirmed in this belief by the observation of the action of three expansion joints in the pipe leading from the central plant to the pump house on the lake, which pipe had been put in before the writer had anything to do with the plant. They have been giving a great deal of trouble by leaking, and they have had to be repacked several times each year; but the pipe is not well aligned nor is the anchorage good, so that it could not be expected, according to the writer's notion, to work well.

All steam and return pipes are covered with a good quality of sectional covering. Various kinds have been used, but it is yet too early to state any positive results as to durability. One thing is certain, however, that the covering must be thoroughly protected by paint, on account of the moisture of the air in the tunnels, and it has been found that the best protection is found by giving the covering two coats of heavy asphaltum paint.

The buildings from which the water of condensation flows back to the central plant by gravity are the following ones: University Hall, North Hall, Law Building, Science Hall, and the machine shop. In the remaining buildings—Ladies' Hall, chemical laboratory, armory, and New Library—receivers are provided, from which automatic pumps return the water to the central plant. It has been found by experience, as should be expected, that these automatic pumps only act as steam traps as long as the pressure of the steam does not exceed the pressure corresponding to the head of water between the pump and the receiver in the central plant plus the friction in the pipe. This limit has made it necessary to introduce traps in some of the buildings where the pumps are used.

All of the buildings connected with the central plant, except the pump house, are now provided with fans for ventilation. In some of the buildings, like University and North halls, chemical laboratory, and the armory, the heating is principally done by the coils in connection with the fan. In the others the fans are principally used for ventilation. In three buildings—Science Hall, Ladies' Hall, and the Law Building—the motive power of the fans is the electric current, whereas in the others steam engines are used, of which there are various makes in use, the exhaust steam being in all cases used for heating. As can be imagined from the fact that a number of different architects have planned the heating apparatus of the various buildings, the systems are of various kinds and efficiency.

During the last five years the heating plants of all of the buildings have been improved in various ways, so that at present all the buildings can be heated and ventilated in a fairly satisfactory manner. The following table contains the amount of direct and indirect heating found in the several buildings. It should be borne in mind that the indirect surface is all used in connection with a fan.

	Direct.	Indirect.
Armory	1,200 square feet	6,000 square feet
North Hall	800 " "	2,000 " "
University Hall	9,000 " "	6,000 " "
Law Building	3,000 " "	1,500 " "
Chemical Laboratory	2,500 " "	3,000 " "
Ladies' Hall	8,000 " "	1,000 " "
Machine Shop	1,500 " "	2,000 " "
Science Hall	8,000 " "	4,000 " "
Pump House	300 " "	
New Library	19,500 " "	3,000 " "
	55,800	28,500

In addition to furnishing steam for the above heating surface, the central plant furnishes steam for heating water in several of the buildings, but especially in Ladies' Hall and the armory, where necessarily a large amount of steam is used for this purpose the year around. Steam is also furnished for experimental purposes to the steam laboratory in Science Hall, and to three engines and a steam hammer in the machine shop, also to the pumps at the lake shore, which pumps furnish all the water for the university and for the State capitol, situated about one mile from the university campus. Only a small part of the university buildings are being lighted by electricity, the current of which is obtained from dynamos run by the engines in the machine shop, the remainder being lighted by gas or electricity furnished by a private company.

The central plant has to be run the year round to furnish steam for the pumps and the engines in the machine shop and steam laboratory. On account of the pumps it is even necessary to run it Sundays during the summer vacation. Steam for heating purposes must, as a rule, be furnished from October 1st to May 15th, and to heat water up to commencement, about June 25th. The total amount of coal used during the year is about 4,500 tons. The kind of coal used has varied from year to year, but has always been of Western origin. Beginning with ordinary screenings—nut and below—it was soon found that because of the high freight charges it did not pay to use such low-grade coal. Then nut coal was tried for a while, but for the last three years washed coal of various sizes, but always smaller than nut coal, has been used and with very good success. The price has varied between $3.25 and $2.30 per ton for the washed coal, the latter price being, however, an exceptionally low one, which probably never will be duplicated. The difference in price at the mine, of unwashed and washed coal (screenings), is certainly not more than

25 cents, and there is no question that with the freight rates which have to be paid at Madison the coal is advanced in value by more than 25 cents by being washed. This has been the result of repeated tests at the central plant, and the writer does not hesitate, from the experience gained, to predict a great future for the washed coal here in the West.

The general efficiency or economy of the plant necessarily varies a great deal. On account of the changeable weather, and also on account of the varying demands on steam from the laboratories and the machine shop, the boilers in the central plant sometimes have to be run very much below their most economical rating, and at other times very much above, in both cases reducing the economy. Boilers are fired up or let out as often as human foresight can see the advantage of a change; but it is impossible to foresee all the demands in time, and the result is as above stated. A large number of tests of the boiler plant have been made, and the writer gives below the results from three of these. No tests have, however, been made in the coldest weather, because of the large amount of work required for wheeling in and weighing the coal. The tests recorded below have been made in moderate weather, and with the aid of a number of students in the College of Mechanics and Engineering. The coal was carefully weighed, and separate account kept of the coal used for the high and low pressure boilers. The tests lasted twenty-four hours in each case, and the code of rules prescribed for conducting boiler tests prescribed by this Society was carefully adhered to, except in the matter of weighing the feed-water. On account of the large amount of feed-water, and because there were separate feed pumps, this was found impracticable, and the reading by meters was substituted. However, these meters were very carefully calibrated in place, the pumps working at their ordinary rate, pumping water of the ordinary temperature, and against the ordinary pressure, the water being weighed and the reading of the meters being taken for several hours. Because of this careful calibration the writer believes that the results obtained by the tests are perfectly reliable. In the tests given below three different washed coals were used, and it is interesting to compare the results. It should, however, be remembered that the results cannot be fairly compared, as they were not the same boilers which were used in all three cases, nor were the boilers run in each case up to the same percentage of their rated capacity, from which

last fact also a difference in economy might be expected. In the first two tests given below, the economizer was used for the high-pressure boilers, whereas it was not used in the third. Again, the first two tests were made two years ago, and the old chimney was then the only one built; whereas, in the last test made, March 10th and 11th of this year, the new chimney was used for all boilers. Because of the better draught obtained with the new chimney, it was to be expected that the results should be the best in the last test, as they actually were. However, it seems also justifiable to draw the conclusion that of the three coals used in the three tests, the New Kentucky coal (Big Muddy) is the best of the three.

Calorimeter tests of the steam from each boiler were made once an hour. Steam-pressure readings were taken every quarter hour. Temperatures of flue gases and draught readings were taken once an hour. Feed-water temperatures were taken once an hour, at the pumps; the temperature of the high-pressure feed was also taken on entering and leaving the economizer in the first two tests. The moisture in the coal was determined by selecting a sample of 100 pounds and drying it for twenty-four hours. The coal used during the first test was Indiana washed nut from Coxville, Ind.; it showed 3½ per cent. moisture. During the second test Illinois washed pea coal from Carterville, Ill., was used, containing 12 per cent. of moisture. This excessive amount of moisture was due to heavy rains the day before and during the test. Washed New Kentucky pea coal (Big Muddy) from Murphysboro, Ill., was used in the third test. It was found to contain about 8 per cent. moisture, part of which was due to a large amount of snow which had fallen on the cars from which the coal was taken.

The boilers used during the first test were those that are numbered 2, 3, 4, 5, and 6 on Fig. 312; during the second test, 2, 3, 4, and 5 were used; and boilers 1, 2, 3, and 4 were used during the third test.

Dimensions and proportions:

	Grate Surface.	Heating Surface.
Boiler No. 1	49 square feet	2,180
" " 2	45 " "	1,550
" " 3	41½ " "	1,185
" " 4	41½ " "	1,185
" " 5	41½ " "	1,185
" " 6	35 " "	956

The economizer has a heating surface of 1,508 square feet.

Test No. 1.

The average pressure on the high-pressure boilers Nos. 2 and 3 was 89 pounds, and on the low-pressure boilers 4, 5, and 6 was 39 pounds. The draught was .177 before entering the economizer, and .43 after leaving the same. The temperature of the external air was 13 degrees Fahr., that of the fire room 72 degrees Fahr., that of the escaping gases before entering the economizer 484 degrees Fahr., and after leaving it 349 degrees Fahr. The average temperature of the feed-water at the pumps for the high-pressure boilers was 172.2 degrees Fahr., and for the low-pressure boilers 183.3 degrees Fahr. The average temperature of the feed-water for the high-pressure boilers before entering the economizer was 171.8 degrees Fahr., and after leaving it 270 degrees Fahr.

The total amount of coal burnt under the high-pressure boilers was 20,338 pounds, and under the low-pressure boilers 24,800 pounds. As the moisture was 3.5 per cent., the weights of dry coal were 19,628 and 23,932 pounds respectively. The total refuse was 1,379 and 1,698 pounds respectively, making 7.05 per cent. and 7.1 per cent. ash. The total combustible was 18,249 and 22,234 respectively. The total amount of water evaporated was 2,623.3 cubic feet, or 159,350 pounds for the high-pressure boilers, and 2,970 cubic feet, or 179,772 pounds for the low-pressure boilers. The quality of steam found from the calorimetric tests was 98.5 and 98.4 respectively. The equivalent evaporation of water into dry steam from and at 212 degrees Fahr. was 169,200 pounds for the high-pressure boilers with the economizer, and 153,200 pounds without it, and 186,450 pounds for the low-pressure boilers.

Water actually evaporated per pound of dry coal from actual pressures and temperatures:

High-pressure boilers	7.997
Low-pressure "	7.48

Equivalent water evaporated per pound of dry coal from and at 212 degrees Fahr.:

High-pressure boilers with economizer	8.64 pounds
" " " without "	7.81 "
Low-pressure "	7.8 "

Equivalent water evaporated per pound of combustible from and at 212 degrees Fahr.:

High-pressure boilers with economizer	9.27 pounds
" " " without "	8.4 "
Low-pressure "	8.38 "

The high-pressure boilers were rated by their builders at 275 horse-power, the low-pressure boilers at 330 horse-power. During the test the high-pressure boilers developed on an average only 204 commercial horse-power with the economizer, and 185 without it; consequently they were run at 32.7 per cent. below rating. The low-pressure boilers developed during eleven hours, when all three were being used, 266 horse-power; during the thirteen other hours, when only 4 and 5 were being used, 190 commercial horse-power; they were, therefore, being run 22.4 per cent. and 24 per cent. below the commercial rating.

Test No. 2.

The average pressure on the high-pressure boilers was 87.6 pounds, and that on the low-pressure boilers 40.26 pounds. The draught was .16 inch before entering the economizer, and .354 inch after leaving it. The temperature of the external air was 38.7 degrees Fahr., that of the fire room 68.8 degrees Fahr.; that of the escaping gases before entering the economizer 356 degrees Fahr., after leaving it 295 degrees Fahr. The average temperatures of the feed-water at the pumps were respectively 159.6 and 196.8 degrees Fahr., and the average temperature of the feed-water at the economizer was 160 degrees Fahr.; after leaving it, it was 237 degrees Fahr. The total amount of coal burned under the high-pressure boilers was 16,238 pounds. In addition 1,867 pounds were used for banking boiler No. 3. The total amount of coal burned under the low-pressure boiler was 20,564 pounds, and in addition 491 pounds were used for banking boiler No. 4.

The moisture in the coal being 12 per cent., the following are the amounts of dry coal used under the boilers:

High-pressure boilers:	14,288 pounds;	for banking.....	1,643 pounds
Low-pressure "	18,099 "	" "	433 "
Refuse under high-pressure boilers.....			1,284 "
" " low-pressure "			1,482 "

making 8.05 per cent. and 8 per cent. respectively.

Total combustible:

High-pressure boiler excluding coal used for banking....			13,120 pounds
" " including " " " "			14,647 "
Low-pressure " excluding " " " "			16,648 "
" " including " " " "			17,050 "

Quality of steam:

High-pressure boiler	98.7 per cent.
Low-pressure "	98.4 "

Total amount of water pumped into boilers and apparently evaporated:

High-pressure boilers	2,008.1 cubic feet, or	.....	122,494 pounds
Low-pressure "	2,348.7 " "	.....	141,413 "

Water actually evaporated, corrected for quality of steam:

High-pressure boilers	.....	120,902 pounds
Low-pressure "	.....	139,150 "

Equivalent water evaporated in dry steam from and at 212 degrees Fahr.:

High-pressure boilers with economizer	.....	131,178 pounds
" " without "	.....	125,617 "
Low-pressure "	.....	144,468 "

Water actually evaporated per pounds of dry coal from actual pressures and temperatures:

High-pressure boilers	.....	7.57 pounds
Low-pressure "	.....	7.58 "

Equivalent water evaporated per pound of dry coal from and at 212 degrees Fahr.:

High-pressure boilers with economizer	.....	8.22 pounds
" " without "	.....	7.87 "
Low-pressure "	.....	7.79 "

Equivalent water evaporated per pound of combustible from and at 212 degrees Fahr.:

High-pressure boilers with economizer	.....	8.95 pounds
" " without "	.....	8.57 "
Low-pressure "	.....	8.50 "

Commercial horse-power:

High-pressure boilers with economizer	.....	158.5 horse-power
" " without "	.....	151.7 "
Low-pressure "	.....	174. "

Builder's rating:

Nos. 2 and 3	.....	150 HP. + 125 HP. = 275 horse-power
Nos. 4 and 5	.....	125 HP. + 125 HP. = 250 "
Horse-power developed by boiler No. 2 for 17.6 hours without economizer	.....	145 horse-power
Percentage of horse-power developed below rating	....	3.33
Horse-power developed by boilers Nos. 2 and 3 for 6.3 hours without economizer	.....	172 horse-power
Percentage below rating	.....	36.3
Horse-power developed by boilers Nos. 4 and 5 for 2.55 hours	.....	135 horse-power
Percentage above rating	.....	8
Horse-power developed by boilers Nos. 4 and 5 for 21.25 hours	.....	179 horse-power
Percentage below rating	.....	28.4

Test No. 3.

During this test the economizer was not used and no boilers were banked.

The average pressure of the steam on the high-pressure boilers was 101 pounds, on the low-pressure boilers 50 pounds. The average draught was .434 inch. The temperature of the external air was 42 degrees Fahr., that of the fireroom 81 degrees Fahr. The average temperature of the gases escaping from boiler No. 1 was 593 degrees Fahr. The average temperature of the feed-water at the pumps was 180 degrees Fahr. for the high-pressure boilers and 179 degrees Fahr. for the low-pressure boilers.

The total amount of coal burned :

Under high-pressure boiler (No. 1)	21,762 pounds
" low-pressure boilers (Nos. 2, 3, 4)	36,090 "

The moisture in the coal was 9 per cent.; therefore the weights of burned dry coal were as follows:

High-pressure boilers	20,021 pounds
Low-pressure "	33,203 "
Refuse under high-pressure boiler	1,925 "
" " low-pressure "	3,193 "

making 10.9 per cent. and 9.6 per cent. respectively.

Total combustible:

High-pressure boilers	18,096 pounds
Low-pressure "	30,010 "

Quality of steam :

High-pressure boiler (No. 1)	97.1 per cent.
Low-pressure " (No. 2)	99.1 "
" " (No. 3)	98.4 "
" " (No. 4)	98.0 "

Total amount of water pumped into boilers and apparently evaporated :

High-pressure boiler	165,993 pounds
Low-pressure "	256,375 "

Water actually evaporated, corrected for quality of steam :

High-pressure boiler	161,179 pounds
Low-pressure "	252,529 "

Equivalent water evaporated into dry steam from and at 212 degrees Fahr.:

High-pressure boiler	172,955 pounds
Low-pressure "	267,767 "

Water actually evaporated per pound of dry coal from actual pressures and temperatures:

High-pressure boiler	8.05 pounds
Low-pressure "	7.61 "

Equivalent water evaporated per pound of dry coal from and at 212 degrees Fahr.:

High-pressure boiler	8.64 pounds
Low-pressure "	8.06 "

Equivalent water evaporated per pound of combustible from and at 212 degrees Fahr.:

High-pressure boiler	9.56 pounds
Low-pressure "	8.92 "

Commercial horse-power:

High-pressure boiler	209 horse-power
Low-pressure "	324 "

Builder's rating:

High-pressure boiler	200 horse-power
Low-pressure boilers, 150 (1) + 125 (2) + 125 (3)	400 "
Percentage of horse-power developed above rating, high-pressure boiler	4.5
Percentage of horse-power developed below rating, low-pressure boilers	14

The difference in evaporative performance between the one high-pressure and the three low-pressure boilers in the third test is partly due to the fact that the former was probably running at its most economical rate, whereas the latter were running at a lower rate. Secondly, the one high-pressure boiler was a Babcock-Wilcox boiler, which had only been used for about four months, and which, consequently, was nearly clean, besides being probably more economical because of its design than the two return-tubular boilers used for the low pressure. It is also probable that the fact that the high-pressure boiler was a larger unit contributed to the increased economy.

With reference to all of the tests it should be stated that no special cleaning or other preparations were made for the tests, so that the results show the actual performance of the plant as run throughout the winter, or at least in similar weather. It is perhaps to be expected that the economy is somewhat higher in more severe weather, when all of the boilers are running more nearly up to their rated capacity, and when the draught, as has been shown by observation, is considerably better than the draught recorded in these tests.

The results, as recorded in these tests, are not extraordinary, but on the other hand they ought to be considered as fairly satisfactory, especially if it is kept in mind that the coals used were all Western, and of small size, although washed.

By various gas analyses it has been found that there nearly always is an excess of air. This was also the case in the three tests. But with a mechanical stoker it is almost impossible to prevent it. In the last test it was less so with the high-pressure boiler, because of the more rapid consumption of coal, and this fact also points in the same direction, as was indicated before, that the boilers will give better economy when run well up to their rated capacity.

In the series of tests, as given above, only the most essential data are given, as it was thought that in a paper of this kind more details would not add anything of interest.

In conclusion, the writer desires to repeat that he does not expect that the plant described is going to be looked on as a model of its kind—quite on the contrary; but because of the peculiarities in the conditions which had to be met, it is hoped that something of interest may be found in the foregoing paper.

DISCUSSION.

Mr. William H. Bryan.—Professor Bull's description of the difficulties met with in the reconstruction of his heating plant, and the methods by which the problems were solved, is quite interesting. The carrying of two different pressures in the same boiler plant, however, is to be avoided wherever possible. It complicates the piping, and involves the use of two independent systems of boiler feeding, besides which more boilers must be kept in operation, as a rule. For instance, when the steam required by each system is a little more than one boiler can handle, three boilers would do the work with the combined system, while four would be required with separate systems. Why would it not have been entirely feasible to have carried 100 pounds on all the boilers, reducing it to 40 pounds for heating? This reduction could be performed by a special reducing valve; or it is quite likely that the ordinary stop valve could be throttled so as to serve this purpose, as the work is no doubt quite uniform. Has this ever been tried?

Professor Bull states that the returns are all discharged into

a receiver through traps, this being necessary on account of the different pressures carried. Would not this be simplified by carrying a pressure low enough to suit the building most difficult to heat, letting its condensation return by gravity, using traps on the others?

Professor Bull also calls attention to the fact that automatic pumps only act as traps as long as the heating pressure does not exceed the pressure against which the pump is expected to lift. In such cases there would be no need of a pump at all, as the returns would flow to the receiver by gravity. If the pressure fluctuates, the pump could be by-passed, so that the returns would flow by gravity whenever the conditions were favorable. Of course, if there is a considerable excess of pressure over that in the receiver, traps would be required. Another good plan would be to apply a vacuum system, such as the Paul or Webster, to the radiators of all the buildings, the returns all going directly to a single closed receiver, those from the low-pressure buildings being pumped.

Referring to smokeless furnaces, my experience shows that the down-draught and other good types are capable of giving good results in smoke prevention and efficiency even when forced considerably beyond their ratings.

Mr. Reginald Pelham Bolton.—There are some considerable questions raised by this paper, which do not appear on the face of it. It appears to me as if it was a very doubtful question indeed whether the decision to employ a central boiler house for this heating work was a good one. The buildings are very much scattered, and the guiding decision, as recited by the author in the paper, appears to have been that a great deal of money had been spent on the boiler house, and that therefore they must spend a great deal more to bring it up to the mark, and a further consideration was that they desired to use soft coal. It would have appeared to me in that connection that, as they were already using a very satisfactory kind of coal without any smoke nuisance, they were jumping out of the frying-pan into the fire by trying to use soft coal, and then adopting expensive devices to do away with the smoke nuisance.

This layout of a central heating station for a number of isolated buildings is a very favorite one. I had one such case, of a university to be heated from a central station, and carried it out somewhat in this manner, not altogether with successful

results, and I am inclined to think that this whole plan would require considerable reconsideration if it were taken up upon a proper basis. I can understand that the author may not have had a free hand. At the same time it should not go on record as being an ideal arrangement. In the first place, the adoption of two pressures for heating is a rank absurdity. It is manifest that it was a concession to the fears of persons who were not at all acquainted with steam practice in any way, and who laid down an arbitrary line as to what was dangerous and what was not dangerous at 30 pounds or 40 pounds pressure! In any case, the carrying of two sets of mains, one for heating and one for power purposes, was absurd, because there were many fans and much other apparatus scattered through the different buildings, the exhaust of which should be used for heating, and which was apparently all thrown away. There is only about 800 horse-power of heating surface in the entire buildings, and they have 1,160 horse-power in this boiler plant, which goes to confirm the idea which I have arrived at from reading the paper —that the exhaust steam is thrown away. The author's conclusion that it would not have paid to have carried in the tunnels steam of low pressure is a false conclusion. I do not see any reason why the low pressure should not have been carried, without reducing valves, to those buildings if it had been necessary; but I would not say that in this case it should be necessary. The best way would be to carry high-pressure steam to all these buildings, to have had the various engines exhausted into the heating system, and to have by-passed the live steam through a reducing valve into the heating system.

The methods by which the condensed water from the heating system was returned to the boiler house were faulty. They appear to have placed receivers with pumps in the various buildings, acting as a trap for the whole of the heating system. That means to say that the steam was put into the radiating system and the return opened into this common receiver, all alike discharging into this one receiver. The consequence was, as stated by the author on page 906, that he found a limit to the efficiency of this receiver and pump as a trap, and had to introduce traps between the heating surfaces and the receiver. That is what I should have expected he would find. The fact is that the whole of this part of the plant might be very greatly improved. The modern practice nowadays is all in the direc-

tion of each heating unit being separately trapped, and that is what should be done in this case.

I notice a very curious statement here, that they added a feed heater, and that the heater, raising the temperature of the feed water somewhat, discharges into a hot-well. It seems to me a very curious piece of practice to heat your water in a closed heater and discharge it into an open well. I should have thought that the proper place for the heater was beyond the well on the way to the boiler. The net result is very ineffective and disappointing. The temperature of the feed is apparently only 170 degrees. It would pay these people to take live steam out of the boiler and heat that up to 208 or 210 degrees, if only for the sake of adding to the life of the boilers.

The author seems to lay a great deal of stress on the statement which appears on page 907, that it has been found necessary to use steam at higher pressure for the tempering coils than is used for the direct radiation under similar circumstances. There is something in that. The tempering coil may be very much exposed to the atmospheric conditions and may need high pressure there; but one would read from that expression of his, and from other parts of his statements, that he uses high-pressure steam in the coils for the fans. I think that is a mistake. Our modern practice shows that you can get as good results from low-pressure as from high-pressure steam. That is a statement which sounds queer, because there are more heat units in high-pressure steam; but it is one which can be borne out in actual practice. The complication of the two pressures which are carried appears to add very greatly to the cost of running this plant. You will see on page 908 that two feed pumps have to be run all the time. There are two mains under pressure, with their friction and their losses, and altogether that is a most undesirable feature, and I should want to change it very promptly.

Passing on to page 908, the author deals with the plant which has been designed for handling the coal. It appears to me that there is a good deal to be said on the question as to whether these coal-handling appliances pay for themselves. I have installed a very large plant of that nature, and now that we have it we find that its chief use is to keep the coal under cover, which might be done at much less expense by erecting sheds on the ground level outside. The cost of maintenance of

these coal elevators is quite considerable, and also the wear and tear upon them. The interest and depreciation, of course, have to be taken into account, and it simply results in having lifted your coal up to a certain point to act as a gravity delivery to automatic stokers. Everybody has his own opinion of automatic stokers, but my experience has been that unless a plant is exceedingly large they are simply a means of making the human stokers idle. The men have to be there, and they stand around and watch this machinery operate. We have to pay them, and I think it is preferable to keep them occupied. As I say, the whole question might be discussed in very special detail as to whether such a plant as this would require coal-lifting devices.

On page 908 I might venture to criticise a small detail in the method of carrying the pipes in the tunnel. I see they are supported by chains with a roller underneath. That is a superfluity; if the chains are there the roller is not wanted, and *vice versa*. Personally I have a strong prejudice against hanging pipes by chains, and always forbid it. There is no method of close adjustment, and the proper way is by a hanger with a thread and nut on it, so that the pipe can be absolutely aligned. By the other method we have no means of alignment.

The author would have contributed much to the interest of the paper, to my mind, if on page 911 he had given us a little more data as to the sectional covering which he used to cover his pipes on the underground work. There is a question which comes in very prominently as to the lasting character of these coverings in such positions, and one would have liked to have learned more about it. I might say that on the question of returns I should have supposed that one of the vacuum systems would have been an ideal addition to this plant. Here is a condition where it is required to get water back from different levels at different points over a large area, and a vacuum system would have delivered that water back without any difficulty. All the independent receivers and pumps scattered about these buildings might then be torn out and replaced by one in the central power house.

A strong point, to my mind, against the adoption of a central power and heating plant in this particular instance is made on page 913. You will notice that the general efficiency and economy of the plant vary a great deal "on account of the change-

able weather, and also, on account of the various demands on steam from the laboratories and the machine shop, boilers in the central plant sometimes have to be run very much below their most economical ratings." That is the whole trouble with these central stations; unless they are very largely subdivided, and you get your units down so that you can be sure of their doing their best duty when the load is light, you get this condition of variation—a large staff of men, very expensive machinery, the whole of which has to be run for perhaps a very small fraction of the total power of the installation. We are in these days a little crazy on central station work, and I think perhaps we are running it into the ground. This is one case in point to my mind. I must not, however, be put on record as being altogether averse to central station plants, for I think they have their proper position.

The tests of the boilers are interesting, but tests of the whole combined plant would have been more interesting, and we then should have learned a good deal more about the value of the entire apparatus.

Mr. William Kent.—On page 914 the paper says: "The moisture in the coal was determined by selecting a sample of 100 pounds and drying it for twenty-four hours." It is not stated how it was dried; but the probability is that it was done in the old-fashioned way by putting it up over the boiler, and consequently it cannot be known whether the coal was completely dried or not. The results apparently showed that the coal was not dried, for the author says: "The coal used during the first test was Indiana washed nut from Coxville, Ind.; it showed 3½ per cent. moisture." It is very doubtful if there is any coal in the State of Indiana which has as little moisture as that. He says further: "During the second test Illinois washed pea coal from Carterville, Ill., was used, containing 12 per cent. of moisture. This excessive amount of moisture was due to heavy rains the day before and during the test." That figure is not excessive. I have found a piece of coal from Illinois that looked apparently dry; it was kept in a dry place under a shed for two months, and had then 14 per cent. of moisture in it. "Washed New Kentucky pea coal (Big Muddy) from Murphysboro, Ill., was used in the third test. It was found to contain about 8 per cent. moisture, part of which was due to a large amount of snow which had fallen on the cars from which the coal was taken." I have

found as much as 8 per cent. of moisture in that same coal when it was apparently perfectly dry. So the moisture tests are probably erroneous.

Prof. Forrest R. Jones.—My work in the University of Wisconsin has been in an entirely different line from that dealt with in the paper. But there are two or three things that I think I might mention here to point out why certain things have been done. In the first place the soft coal is very much cheaper out there than the hard coal, and that was the main reason for using it. In all the furnaces which are in use there is very little smoke, so there is no objection from this point of view. As to its being an ideal or model system, the very first few lines of the author's paper say that it is not a model one. And, as for having sheds on the ground, the regents of the university are justly proud of the beauty of the campus. They have there what they believe, and what I believe, is one of the most beautiful grounds of any university in this country, and I think there is only one university whose grounds equal or surpass it, and that is Cornell University. They desire just as much to keep their grounds looking handsome as residents of Washington desire to keep Washington handsome. For that reason they have gone to some extra expense in many cases to prevent the erection of any building which would detract from the beauty of the grounds. For this reason no sheds for coal, or anything of that kind, have ever been erected on the grounds. Great care has been taken to keep all unsightly structures off the ground, and this was one reason for using the underground-storage bin for the coal. As it is now arranged its roof is flush with the surface of the earth. The coal is hauled up, dropped down through doors in the roof, as commonly done through a sidewalk, and is out of sight, and there is nothing unpleasant to the eye about it. Another thing, by keeping the plant as a central plant, they have only one place to which coal must be hauled and from which cinders must be taken away, exclusive of what we call the agricultural buildings, which are about half a mile away from the central heating plant, and therefore the wagons hauling coal and cinders go into only one place on the grounds—near the edge—and they do not have to pass over the driveways, which are kept up in the best condition throughout.

Mr. Robison (Bureau of Steam Engineering, Navy Department).—There is quite a long distance between a college in

Wisconsin and a naval engineer's problem. But the subject of a central power station such as they have in Wisconsin University is one which excites the naval engineers just at present. The steam-engineering machine shop in New York was destroyed by fire last winter, and the Engineer-in-Chief, the President of the Society, is at present getting out plans for a new shop. The subject of a power house for that shop has come up, and in this connection has come up the question as to whether or not there should be for the navy yard at New York a central power house, which should supply electric power to all the various machine shops, boiler shops, etc., in the navy yard. In order to understand the question that we have there, it is necessary to consider a little bit the conditions of navy-yard work. There are at the New York yard, I think, four or five machine shops, two or three boiler shops, foundry, pattern shops, etc. It would naturally be best to have only one shop of each kind, and I think that about fifteen or twenty years ago Secretary Chandler gave an order which would lead to such a thing. But the bureau system, which is fixed by law upon the navy, and the appropriation system made that single shop uneconomical; it cost more for the clerk hire and for transfer of accounts than you saved in the labor at the machine shops. In the navy yard there are about forty or fifty buildings, some twenty of which require power. I think the power is given to those buildings at present from five central power stations. The point is whether or not we should have only one electric power station to supply all of these buildings, or whether we should take one power house which would supply each separate group of buildings with steam for the hammers, for heating, and, in the case of the steam-engineering building, for a test boiler. The central electric power station presents, of course, advantages in the matter of the cost of an electric unit. I do not know that when the amount of power required in the various buildings is as great as it is in the case of the New York Navy Yard you are going to save much in the cost of the unit of electricity. You have got about 700 horse-power. You can get it about as cheaply, if you generate it where you are going to use it, as you can get it from a distance of half a mile with the larger units used in the central station, using large enough units in a 700 horse-power installation to get economical results. I think that the cost of operation in the two cases

would not be much different. The cost of installation would be reduced of course. In this connection money costs the Government just 3 per cent. You have to consider that when you are considering any question of a Government plant. It is not a case of 6 and sometimes 7 per cent. for commercial institutions; it is a case of the 3 per cent. that money is worth to the Government, and, in my opinion, any questions of economy should be based upon the rate at which the Government can borrow money. It seems to me, speaking personally only, that where you have got to have a steam plant of any size whatever, as we have to have in the New York Navy Yard, it is just as well to include in that steam plant your power house. You save the cost of transmission, and that is equal to at least the amount of saving that you would have in the use of the somewhat larger unit which you get in a central station, and you save also some of the difficulties connected with heating and with transmission of steam to considerable distances for steam hammers, etc. We have to have in the steam-engineering shops steam of 450 or 500 pounds pressure for testing the valves and pipes of steam work, and as long as we have to have a steam plant of any size whatever we are not going to save enough in labor, if we have a central station, to make it pay. There are some of the other plants there, though, where I think that it might pay. The Bureau of Ordnance has a plant up there that I think has about 125 horse-power. The Bureau of Equipment has one of only 25 horse-power. This could readily be run by electricity, and economically too. They would save money probably. But I do not think that if you had plants of the size which ours is going to be—some 1,200 horse-power—that you would save much money. Perhaps you could operate steam hammers with compressed air, in spite of some difficulty in the exhaust passages of those engines; but, at any rate, we have to have steam in the steam-engineering building for our testing boiler. We ought to have steam in the buildings themselves, I think, where the distances from a boiler house are as much as half a mile. I do not believe in transmitting steam half a mile. You lose too much. The President of the Society, the Engineer-in-Chief, who has this work in hand, has directed me to request a discussion of the relative merits of the two systems which could reasonably be installed at the New York yard: first, a

central power station, which would be under the control of the Bureau of Yards and Docks, this bureau already having charge of the central lighting station; and, second, a system of power stations, each to supply the buildings in the immediate vicinity of the station and control the supply for the bureau under whose control this separate station would be. There are some particulars where it would pay, without question, to have a central station. For instance, we have to use pneumatic appliances in navy yards to-day. They save money, and, while money is only worth 3 per cent., economical administration is the point which is desired by at least one, and probably by all, of the bureau chiefs. We have got to put in, if we want to use pneumatic appliances, a separate air pipe, running all round the yard to the different docks, and the Bureau of Construction will need separate air pipes too. It seems to me foolish to have three or four lines of pipe doing the same work when you can have one pipe doing the same thing. That looks foolish to all of us. It seems to me that the Bureau of Yards and Docks, which is charged with the administration of the yard, maintenance of the means of transportation, material, stores, should put in a line of piping for pneumatic or hydraulic pressure for necessary transmissions. It might probably and properly take charge of electric transmission as between buildings. But I do not think they ought to have a central power station to attend to the whole thing. As I said, gentlemen, the Engineer-in-Chief asked to have a discussion of this question to invite the opinions of the members as far as possible.

Prof. W. S. Aldrich.—I heartily sympathize with Professor Bull of the University of Wisconsin in this matter, reciting his experience along what might be termed the engineering end of a great university. The State universities of to-day are undergoing a radical transformation, and there is no telling when the end will be reached or a condition of stability attained, not only in business management, but in engineering education as well as in engineering economics. We have here an account of the efforts put forth in that direction in one institution.

It is quite evident that there has been brought about in this university power plant a condition of affairs not altogether ideal. Harmonious development may be had from the drawing board in laying out the project for a tall office building or a new university, *de novo*; but one would not look for it in the line of

development by the method of accretion, adding to and laying on from the outside, rather than in pursuance of any distinct organic law of growth and development from the start. It is an exceedingly difficult matter, with changing politics and policies, with changing governing boards and administrations, to maintain any definite line of economic policy along the engineering features of an educational institution. Such an institution requires light, heat, and power. You will supply those three things, it is often thought, at the lowest rates when you have centralized the plant; but it appears from this paper that other equally important features have to be considered.

When some of these buildings were installed the ventilating appliances for mechanical circulation of air were operated by steam engines. These called for a certain steam pressure different from that for the heating service. All understand well enough without discussion the great disadvantages of handling two steam pressures quite as much as in handling two electric pressures.

As far as Professor Bull has here noted, there are at least twelve little steam-driven auxiliary engines throughout the plant of the University of Wisconsin. These auxiliaries have come to be as they are and located where they are by reason of this gradual accumulation of engineering installations in the different buildings. Part of this is, no doubt, owing to the fact that architects specified certain particular systems of mechanical heating and ventilation for the buildings. Now, it is quite apparent that the axiom underlying a large part of this question of auxiliary steam engines is that their non-expansive working is expensive working. Some of these auxiliaries are quite likely to consume from 150 to 350 pounds of steam per horse-power per hour.

Such a rate of steam consumption for the five small oscillating steam engines for the mechanical stokers, the two low-pressure feed pumps, the one duplex pump, the one duplex air pump, the one small engine for the coal conveyor, the one engine for electric lighting, and the several small engines for heating and for ventilating plants, will clearly demonstrate that where we have attempted to save money in a university by putting in a central steam-heating plant we have actually introduced another element of very great wastefulness in the use of steam for the necessary auxiliaries.

It is a small percentage, perhaps, but figure it up and it will be seen that that small percentage—a little drop in the bucket it may be—is going on continuously with the operation of the plant. While you are saving money at one end of the line you are wasting it perhaps almost as fast at the other end of the line. In other words, since we must have electricity, it would appear best to generate enough electricity to run all of these auxiliaries. The point is well taken: if we are going to have power, have it in abundance, generate it in large units, save money by its judicious application. When we have electricity for lighting, and there is a dynamo running continuously, it would be good engineering to put in a somewhat larger dynamo unit and operate all of these auxiliaries in the university from the central generating plant.

Another thing here brought to our attention is the question of the location of power plants. Engineers are blamed for almost everything with which they have anything to do, from the least unto the greatest. But there are certain things that engineers cannot control in State universities, when they find themselves placed in charge of engineering departments, and that is the location of their department buildings. However aggressive they may be, however pushing and energetic, the location of department buildings in almost all State universities is so controlled by the policies of governing boards that it is impossible to bring to bear any engineering argument which will appeal unto them. An illustration is given here of the use of high-pressure steam—the very word high-pressure being repellent; danger lurks within that. The idea of locating the boiler house of a power plant in a mechanical-engineering department on top of a hill is not new. There are other institutions which have entertained the same idea. Of course, it is not necessary to go into details, but we know that what has been done in one institution has been going on all over the country. The federal endowments by which these engineering departments are largely maintained are developing in every State and Territory similar schools of engineering and the mechanic arts. In course of time, gathered about these engineering departments, will be the light, heat, and power plants of the university.

Mr. Francis H. Boyer.—It is unfortunate that engineers have to come in contact with the cashier or treasurer of large corpora-

tions. If we could go ahead with our engineering ideas and devices and experiences which we get here it would be a great thing. You take these large corporations where the question arises of centralizing power plants or even engine locations, and where, should a serious accident occur which would stop the entire works, the economy offered by centralizing is sure to be offset by the risk of accident. I believe the most remarkable case of installation for power purposes, for heat and light and power, that we have in this country is our street railroads. Take, for instance, the street railroad of Boston, with which I am somewhat familiar. They have an installation probably of about 30,000 horse-power divided into seven units, if I remember correctly. They are all run in multiple together, and there come times in stormy weather, when we have intense sleet, that a station is thrown out, and instantly the work of that station, which may be 1,200 or 1,500 horse-power, is thrown on the remaining stations. I have stood at their gauge boards and watched those processes during heavy storms, and it really is a terrific sight. Suppose they had their entire work on a central power station, which is feasible; under these severe cases they would become inoperative—their business would stop. Take these large refrigerating plants which we have over the country —Squires', at Boston, Armour's or Swift's, at Chicago—and suppose they had everything there confined to one plant, and a fire destroys the engine department. With the large number of buildings which they have, they must have plants here and there which in case of an accident can help the other. There is where we engineers rub against the treasurers, and against the chances which they take, and it is a point that we cannot ignore.

Mr. Wm. Kent.—I am inclined to favor one central station if the distances are not over half a mile. I do not know but that I would go up to a mile. I have no fear of carrying steam that far. It is only a question of putting enough money into covering the pipes so as to reduce radiation. One plan for reducing the radiation where there are variable loads is to have two separate steam pipes, one smaller and one larger; one, say, 6-inch and one 8-inch, instead of a 10-inch pipe. In summer time use the 6-inch, in fall the 8-inch, and in winter use both.

I would suggest another plan for saving the heat of condensation in long lines of steam pipe and for facilitating the operation of a central power plant: have a tunnel underground leading

from the power station to where the steam is to be utilized, large enough for a man to stand upright in it, say, 6½ feet high and 2½ feet wide. In that put the steam pipe or steam pipes, the returns, the pneumatic pipes, electric wires, and everything which has to be distributed. Have the boiler house run on the induced draught system, so that all the air which gets into the grate bars is pulled over by the fan. Now open this tunnel into the boiler house so that there is a continual circulation of air through the tunnel from its outer end. If there is any radiation of heat from the steam pipes warming the air in the tunnel, all that heat, except what is radiated through the brickwork of the tunnel, will be swept back under the grate bars and used over again. That need not be the only supply of air. There may be some additional holes in the wall of the boiler house to let air into the boiler house; but have a small, gentle circulation of air carried through that tunnel from the extreme end to the boiler house and save that heat thus wasted, and put it under the grate bars where it belongs.

Mr. W. S. Rogers.—I have always been a "central station" fanatic until the recent winter's storm in Boston. Having seen the advantages arising from one station helping out another during breakdowns, as has just been explained by Mr. Boyer, and realizing that I would have been snowbound if it had been a central station plant, I have become completely converted to a system of divided stations. But if engineers must have central stations let them duplicate the first one and have a relay.

I do not advocate the use of a central-power station in a university using a main generator and electric transmission for driving ventilating fans and other instruments in use about the building, for I do not think it wise to dwell too much on economy in the running of our universities.

I have found in my experience that every time we "waste" money to educate a man we have saved barrels of money afterward. There is no waste of money in education, and were I running a university I would obtain old oscillating and rotary engines and keep them going along with the modern types that the students might learn how not to do it as well as how to do it. And when they go into the old plants with the old types of machinery they will realize the good side of a dollar in profits and make the old scrap earn enough to justify the purchase of

modern machinery. Then they have treasurers and stockholders favoring their plans.

Prof. Forrest R. Jones.—I agree perfectly with the last speaker in regard to teaching students what not to do. But we try generally to put in good apparatus to do the work which is required in what may be termed a commercial manner. Experimental apparatus is used when it is necessary to show a man what is wrong. I might say in explanation of this university plant that, outside of the experimental apparatus, the electrical plant consists of two small generators, one an arc machine, mainly for lighting the shops, and the other a very small direct-connected 110-volt direct-current machine for furnishing current for experimental work in the physics laboratory and for projection lanterns throughout the university during the day. The 110-volt generator runs all day. A great variety of other electrical machinery is used for experimental purposes. The experimental machines run only when the students are in the laboratories. As to having a central plant, the regents asked the State Legislature, which has been in session during the last two or three months, for $35,000 to put in a central electric plant to furnish light and power for the university and light for the State capitol, but for various reasons it was decided best not to put in this plant. Just what those reasons are would be a little difficult to get at, perhaps, but it is a State university; therefore politics and the interests of others may enter in some degree. So it was voted not to have the plant, although the Legislature was very generous to the university otherwise, giving them a considerable amount of money for an engineering building and other purposes.

Mr. William J. Baldwin.—When an engineer is called in in a matter of this kind (central plants versus individual plants), he should come unprejudiced. He should have no prejudice in favor of any particular system until he has made a thorough study of the problem before him.

Take a university or navy yard (both having been mentioned here). We must view the whole question, and view it for both winter and summer, before we can arrive at a proper conclusion, as what may give the greatest economy at one time of the year may be very wasteful at another. What is required to be determined is the greatest economy for the whole year. This is the first consideration. Take a navy yard, for instance, and

assume that you have a central steam plant; the question is, Do you want the central electric plant also? In my hasty judgment, you do not, though this, of course, is open for deliberation.

If you have a central steam plant, you can carry your steam through small pipes at high pressures, and get steam at any point of distribution, almost without regard to alignment.

If you have a central electric plant in connection with the central steam plant, you will have to make all the exhaust steam at one point, in which case you will be obliged to use very large pipes to carry exhaust steam to the points of distribution, *i.e.*, the points where you want to warm the buildings, and the question of alignment and drainage (water separation) will then become of the utmost importance.

In all central electric-light plants of New York and other large cities the exhaust steam is thrown away—absolutely lost—and representing what may be called a large fortune, and for the present, in cities, it appears this cannot be helped, as the right of way for large low-pressure steam pipes is practically out of the question.

Take a group of university buildings, and assume they have a central steam plant under one roof. The coal can be hauled, stored, and cheaply handled; you can economize in boiler-room force, and the products of combustion can be distributed or held together at some point where they will least affect the buildings, and this is very essential where soft coal is used.

From this central plant you can take high-pressure steam to your individual buildings, there run your steam engine, and generate your electricity for lighting or power, and produce the exhaust steam on the premises, where, of course, it can be used at once for heating in winter, or allowed to run to waste in summer. When it is condensed in the heating apparatus, of course you can return it by automatic pumps from each local position, and no special attention need be paid to the alignment of the return pipe through the ground to the central station.

As a matter of fact, the amount of exhaust steam left after the elevator service and the lighting a building, at least in the big buildings of New York and other places, is almost always sufficient to warm the building.

I notice, however, that there are men here in favor of the central electric plant for navy yards. I am not sure that there

will be economy in this, taking it as a whole. There is much machinery to be driven in the shops, and comparatively little lighting done, except in the short days of winter. Several of the speakers evidently are considering the question of lighting alone, and are not considering power to any great extent, if at all. In the winter time, when you desire to run engines which will make exhaust steam, the exhaust steam should be where you want to use it; and, as I said before, there is no difficulty in taking the condensed water back to the central steam station, whether up hill or down, and I am rather of the opinion that the hill after all is the place to put the steam station; although this has not been the custom. We generally place the boilers in the valley so that we can carry the steam up hill. This, I believe, is a mistake, for more reasons than one. When the boiler is in the valley, the course of the live steam is generally up the hill, and the tendency of the water carried into or formed in the pipes is to move in the contrary direction to the flow of the steam; whereas, if the boilers were on the hill, the flow of steam would be down hill, and the flow of water would be in the direction of the flow of the steam, both travelling in harmony. Moreover, when you have the boilers on a hill the chimney can be carried above all the surrounding buildings, so that the smoke and gases can be carried over them, whereas with the usual methods the chimney is generally just high enough to let the smoke flow across the campus and into the windows of the buildings when the wind is favorable.

With regard to carrying the condensation from the buildings to the central station, it makes little or no difference where the central station is. The power to send the water back must be at a point where condensation takes place. Usually the water must gravitate to the lowest point in the building, and must then be taken out by pumps, and sent up the hill, if there is a hill.

The fact of distributing the steam at somewhat various pressures to the different buildings has very little to do with the result. When a little error is made in proportioning pipes, it does not make much difference, with a high-pressure central plant of, say, 125 pounds pressure of steam at the boilers, and reducing valves at the buildings, to cut it down to 5 or 10 pounds for heating, and to 100 pounds or thereabouts for engines.

Therefore, it looks to me that the central high-pressure steam plant with comparatively small pipes and distributed engine plants will secure the greatest economy in a navy and also in a university group. Of course, where the buildings are very close together this may be modified.

Mr. Paul H. Grimm.—I think this question of establishing a power plant at the Brooklyn Navy Yard should be considered from a standpoint different from that of the central power station, or a central electric station situated on high ground; we must bear in mind that the Brooklyn Navy Yard is located very close to tide level. Hence I submit that a tunnel, such as Mr. Kent speaks of, for the accommodation of steam pipes and electrical conductors, would be impracticable; then, again, the distribution of power in a navy yard should be considered more from the standpoint of a large manufacturing concern with buildings covering a large area rather than a street railway central station.

The navy yard has groups of buildings scattered over considerable territory, and it seems to me that if power units suitable to these several groups were established convenient to these buildings there would be great advantage at times to be able to run the machine shop, boiler shop, or perhaps some other building, at night in an emergency without interfering with any of the rest of the plant. It may be said that there is greater economy in one central station in the matter of labor-saving, and this, I think, is the only point that could be claimed in favor of the central station plan in this case, but in many instances, especially in large manufacturing concerns, where the question of operating the whole or any part of the plant at all times is of prime importance, the question of labor-saving often makes a very small offset against the other advantages of the separate unit plants, and I think this would be especially true in the case of the navy yard. I believe that a number of units established reasonably near the point of consumption of power would be preferable in this case to one central station, no matter how well equipped, nor how well arranged for economy, especially when it contemplates carrying the steam and electricity from that point to all the different points of consumption in tunnels.

*Professor Bull.**—After reading the discussion, the author of

* Author's closure, under the Rules.

the paper feels especially sorry that he was unable to be present at the meeting when the paper was discussed, as he is perfectly certain that various misunderstandings in regard to the paper, under which several of the participants in the discussion labored, could easily have been explained, whereby a number of the criticisms would have been shown to have no foundation in fact. As it is, it will be necessary to take these up in detail, and, for convenience' sake, they will be considered in the order in which they appear in the discussion.

Mr. Bryan's statement that the carrying of two different pressures in the same boiler plant should be avoided wherever possible, I fully agree with. However, at this central heating plant it could not be avoided for several reasons, the principal one being that several of the boilers on hand when the plant was established could not carry as high a pressure as 100 pounds, which was the lowest pressure that could be used for power purposes at the university. If these boilers could have been disposed of and stronger boilers substituted, no doubt a decided economy and a great simplification would have resulted, but such was not the case, and the necessity of carrying two different pressures is one of the several shortcomings which prevented the plant in question from being a model one, as repeatedly stated in the paper. The above explanation also answers Mr. Bryan's suggestion of the use of one reducing valve at the boiler house.

The pressures carried on the heating mains in the various buildings vary from day to day, depending on the caprice of the janitors who take care of the buildings. The steam pressure will therefore not be the lowest in the same building at all times, which fact makes it necessary to provide the returns from *all* the buildings with traps, and not, as suggested by Mr. Bryan, all but the one which is most difficult to heat. I suppose that Mr. Bryan means that the pressure carried should be *high* enough to suit the building most difficult to heat, and not *low* enough, as stated in his discussion. It might also be pointed out at this place that four of the buildings heated are below the level of the boiler house, so that the return water *must* be pumped back and could not return by gravity, as suggested by Mr. Bryan in the same paragraph.

Mr. Bryan does not seem to understand my statement in regard to the non-working of the automatic pumps as traps.

These pumps with receivers are only used in the four buildings which are lower than the boiler house. Suppose the pressure carried in the building be 10 pounds, then, if there be no traps, there will be the same pressure in the receiver, and if the resistance to the flow of the water back to the boiler house, including the resistance due to the difference of level, be smaller than the 10 pounds carried, the water will pass through the pump, without its working, and there will be a pressure of approximately 10 pounds in the receiver at the boiler house, which will prevent the return water from coming back from all of those buildings in which the steam pressure is lower than 10 pounds or thereabouts, because the traps in these buildings will not work. As it is arranged now in the boiler house, there is no pressure in the receiver, so that the traps will always work whatever be the pressure carried in the various buildings. The use of a vacuum system throughout the whole plant would be a very good thing, but it would not do away with the traps, as it has been found necessary to use steam of higher pressure in the coils connected with the fans than is used in the direct radiation or which would be used in case a vacuum system was introduced.

I did not intend to raise the question of the advisability of the establishment of the central plant in my paper, but quite a large part of the discussion was directed at this question, and I therefore also take the liberty of contributing my share to this discussion.

Mr. Bolton seems to think that it was a very doubtful question whether there was any advantage in the establishment of the central heating plant, as the circumstances were at the University of Wisconsin. But it seems to me that the reasons why he doubts the wisdom of the move are entirely false. First of all, the establishment of the central plant was not owing to the fact that a good deal of money had been spent on the boiler house, and that therefore more must be spent to improve it. If Mr. Bolton will read what is said on the first two pages of my paper, he will see that it was only the *location* of the central plant which was determined by the circumstances mentioned above. In the second place, Mr. Bolton's statement that we were jumping out of the frying pan into the fire by trying to use soft coal in place of hard coal proves to me that he does not appreciate the circumstances under which we in the

West have to labor. The price of hard coal to the university has in late years varied between $5.50 and $6.50, whereas the coal which we have been using in the central plant has cost us from $2.30 to $3.25. This coal has given good satisfaction and has cost on the average less than one-half of what the hard coal would have cost. It ought to be unnecessary to state that, notwithstanding the adoption of "expensive devices to do away with the smoke nuisance," as stated by Mr. Bolton, the soft coal is very much cheaper to use than hard coal—here at Madison. Some of the reasons for the establishment of the central plant have been given by my colleague, Professor Jones, but I will state that the three principal ones were economy in heating the buildings, reduction in danger of fire, and increase of cleanliness, all of which expectations have been amply justified by the results obtained.

That there are faults in the layout of the plant I do not doubt for a minute, but Mr. Bolton must have entirely overlooked what I have stated in the paper, that the plant is not a model one, for he says: "At the same time it should not go on record as being an ideal arrangement." It may be that the carrying of two pressures was "absurd" in the mind of Mr. Bolton, but it was a necessity, as explained before; consequently an absurd necessity.

It was explained in the paper that there were two mains leading to each building heated, one large one for use during the cold weather, and the other for the milder weather during spring and fall. Steam of both pressures is *not* carried to these buildings, with the exception of the machine shop and the basement of Science Hall, where steam has to be used for power purposes, and in which buildings the heating is done by the steam of the lower pressures. The higher-pressure steam is alone taken to the pump house, where the necessary heating is done by this steam, the pumps being driven by condensing engines. Mr. Bolton has also concluded—but without good reason—that the exhaust is being thrown away. All the exhaust from the various engines driving the fans, pumps in the various buildings, as well as from the engines in the machine shop, is being used for heating during the daytime, but, as these engines and fans do not run during the night, live steam from the boilers must be used for more than half the time. The exhaust from the engines in the experimental steam laboratory

for obvious reasons cannot be used for heating, so that there is nothing wasted, and Mr. Bolton's advice of how to utilize the exhaust steam is entirely superfluous. Mr. Bolton's expressed opinion, that it would be entirely practical to carry low-pressure steam in the tunnels, has not been contradicted by me in my paper. I only stated that it would be better to carry high-pressure steam and reduce the pressure at the buildings, and my statement that it would not pay to carry low-pressure steam is not a "false conclusion." On the contrary, Mr. Bolton himself admits its correctness by saying that the best way would be to carry high-pressure steam to the buildings. I suspect that Mr. Bolton has misunderstood me in thinking that I meant to state that, if high-pressure steam could not be used in the tunnels, it would be better not to have a central heating plant at all—which, as explained above, I have not stated.

Mr. Bolton also states that "the whole of this part of the plant (referring to the returns) might be very greatly improved. The modern practice nowadays is all in the direction of each heating unit being separately trapped, and that is what should be done in this case." If the reader will refer to the tenth page of the paper, he will see that it is expressly stated that all "the return water is being discharged through traps into the returns," and, further, "that the heating apparatus of each building is entirely independent of that of every other building," a sufficient answer to the criticism quoted above, the making of which can only be explained by a very careless reading of the paper.

The criticism of Mr. Bolton in regard to the location of the feed-water heater is rather to the point. However, because of special reasons, which it would take too long time to enumerate, it was found impracticable to place the heater between the feed pump and the boiler, and I will only state that the feed-water heater is of small importance in this plant, as the amount of exhaust steam available in the boiler house is very small compared with the amount of feed water to be heated, and it would not nearly heat the water up to 170 degrees, which is considered so disappointing to Mr. Bolton, even if the heater had been placed in the proper position relatively to the pump. In the paper it was explained how this temperature of 170 degrees was reached.

I am very certain that there is no statement in my paper

which justly could lead Mr. Bolton to believe that I used "high-pressure" steam for the tempering coils. On the contrary, on the eighth page, I state that it is desirable to use a "higher pressure" for these coils than for the direct radiation. As a matter of fact, it may be stated that the pressure is hardly ever higher than 10 pounds for these coils.

As to the advisability of using coal-handling machinery, there may, of course, be a variety of opinions. However, my experience with this plant during these five years has fully convinced me that the machinery is a very well-paying investment, taking everything in consideration. It might also be pointed out that a very important point in this particular instance is the cleanliness resulting from the use of coal-handling machinery in connection with the mechanical stokers.

Mr. Bolton has probably forgotten that the primary reason for using the stokers in this instance was the prevention of smoke when using bituminous coal; otherwise he would not have stated that "they (the stokers) are simply a means of making the human stokers idle." I desire, however, also to go on record as being of the opinion that these mechanical stokers are a good investment in other respects besides the prevention of smoke. They are clean, allow the use of an inferior grade of coal, and reduce the number of the required "human stokers." Against these advantages there are, of course, disadvantages, which, however, in my opinion, do not balance the first.

The criticism in regard to the supporting of the pipes in the tunnels I appreciate, and I am aware that the suspension of the pipes by means of chains is not an ideal one. On the other hand, the method advocated by Mr. Bolton seems to me entirely impractical in a tunnel; whereas the other and better methods which have been used at other places are so costly that I could not use them in the plant described. We have found it possible to align the pipes in the tunnels very satisfactorily by means of the chains, a proof of which is the proper working of the numerous expansion joints.

Mr. Bolton finds an argument against the adoption of a central plant in a statement in my paper to the effect that the efficiency of the plant varies a great deal on account of the variation in the consumption of steam from time to time. That this is true does not need any proof, but I must insist that the conclusion which Mr. Bolton draws from the statement is

entirely erroneous. If there were no central plant, then in mild weather, for instance, it would be necessary to run one boiler in each building, but very much below the rated capacity of each, and therefore very uneconomically, whereas in the case of the central plant less than half of this number of boilers would have to be run, and more nearly at their rated capacity. As to the variation of the consumption of steam during the various hours of the day it is only necessary to state that, although the total variation would be greater in the case of the central plant than for each of the separate boilers, yet it would be smaller for each boiler in the central plant than for each of the separate ones, and consequently the boilers in the central plant will be run at a more economical rate than the boilers in the separate buildings. It seems to me that one of the strongest reasons for the establishment of a central plant is the fact that the boiler capacity can be more nearly adapted to the consumption of steam than when the boilers are located in the separate buildings.

Coming to Mr. Kent's criticism of the results of the testing of the coals used for moisture, I must admit that he has good reason to be sceptical. The method used was the old-fashioned one of placing the coal on the boiler, or rather on the breeching. The statement in the paper that the drying was done in twenty-four hours is a slip—the coal was left so long on the breeching as the weight was decreasing. I fully realize that the percentages of moisture in the three kinds of coal tested and given in the paper are not absolute – only relative.

Professor Aldrich's general criticisms of the wastefulness of the many small engines and pumps scattered about in the various buildings I agree with to a certain extent. But if he remembers that all of the steam used by these engines is utilized for heating afterwards, I think he will agree with me in placing the ultimate loss due to the wastefulness of the engines as very small. I suspect that Professor Aldrich's experience with respect to "the changing politics and policies" of the administration has been a great deal worse than mine—otherwise he would not have brought this side out so prominently. I can state with utmost frankness that politics has never had anything to do with the administration of the University of Wisconsin, and the changing of policy is nothing else than one can expect in any corporation.

DCCCXXIII.*

ROLLING-MILL FLYWHEELS.

BY JOHN FRITZ, BETHLEHEM, PA.
(Member of the Society and Past President.)

In the early days of rolling mills the machinery was of the crudest character. The men who built and looked after it were called millwrights, or forge carpenters; consequently, wood was used in its construction to as great an extent as was possible. All of the earlier flywheels, as you will see by the tracing, had wooden arms, at least so far as I know, and Fig. 319 shows the general way of fitting them in. They were made of hardwood, locust or white oak preferred, and about 6 to 8 inches in thickness, width as per tracing.

Figs. 316, 317, and 319 give the plan of securing the rims of the wheels together. The segments are in halves, and secured together with gibs and keys, same as shown in Fig. 319. Having no machine tools when these wheels were built, they were put together practically as the castings came from the foundry, with the lumps and swells chipped off with a two-handed chisel and sledge, which naturally caused a weak wheel. They were made of cold-blast charcoal iron, which was greatly in their favor, and no doubt this fact prevented many accidents.

In the course of time the millwright and carpenter were displaced by the machinist, and with the introduction of machine tools, especially the planer, the wheels were better fitted up, but not much safer, in consequence of the use of iron made from mineral coal, in place of charcoal.

The machinist now became the mechanical engineer, and, wood not being his forte, he made the flywheel all of cast iron, as shown in Fig. 318. The centre, arms, and segments were cast separately, and had the ends of the segments been well fitted together the wheel would have given no trouble. The slots for the tees were, unfortunately, too close to the end of the segment; the result was

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

that they pulled the end of the casting off, and we had to chip grooves across the face of the wheel, and put heavy links around them, as shown on tracing, Fig. 318, which made a good wheel out of it. This was done in 1854, and I think the wheel is yet in use, but never ran at high speed.

Not considering these wheels safe, I was in constant fear of their going to pieces, which is generally attended with the most disastrous results. I abandoned entirely the old mode of making wheels, and adopted plan No. 5—casting the centre and arms together, and the rim in one piece, with lugs on the inside of the

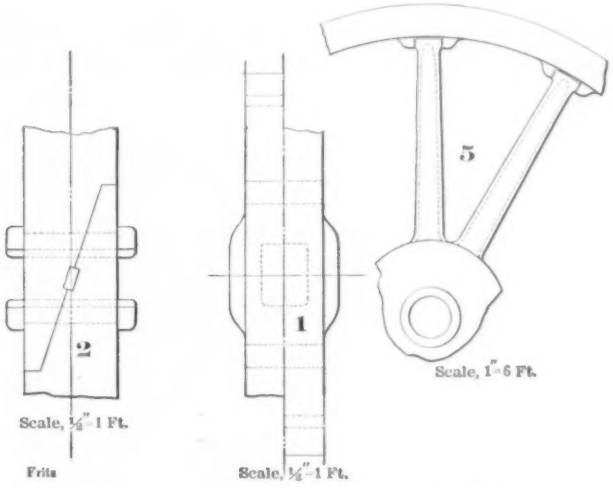


Fig. 317. Fig. 316. Fig. 320.

rim, as shown on tracing, Fig. 320. On the ends of the arms were pads to correspond with the lugs on the rim. When the arms were put in the rim there was a space of about one-quarter of an inch, in which oakum was driven tightly, which kept the arms in place. This made an excellent wheel, and in some cases wood was used to fill the space in, which thin steel keys were driven. There are several of these wheels that have been in use over thirty-five years, and, so far as I know, never have given any trouble. The wheels were 20 feet in diameter, with rims about 12 inches square.

With the introduction of steel for rails and structural material, it became necessary to increase the diameter of rolls. This change necessitated heavier wheels, and larger in diameter,

which made it impracticable to use solid-rimmed wheels; consequently, some plan that was portable had to be adopted. After much thought the plan represented by Fig. 321 was concluded the best and safest, and there are now a number of wheels of this type in use, varying in diameter from 20 to 30 feet. I do not know that any one of them ever has given a particle of trouble in any way, and there are a number of wheels, as per Fig. 321, that have been in use for twenty-five years, some which have encountered, and are daily liable to encounter, the most severe strains that it is possible for a flywheel to be subjected to, and are to-day apparently as safe as the day they were started.

It has been said that flywheels go to pieces in consequence of the irregular work, and engines driving dynamos for power

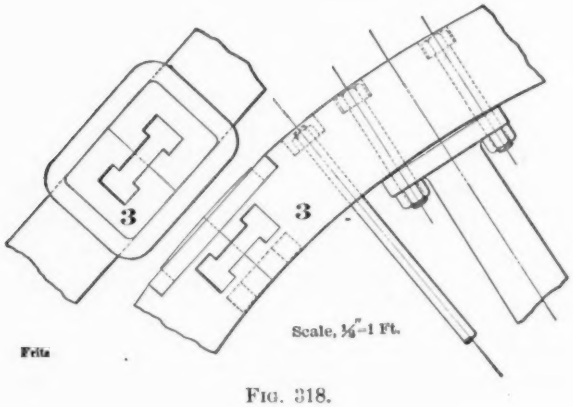


FIG. 318.

purposes have been included in this class; but the strain on a flywheel driving a dynamo cannot be compared with that of a rolling mill. In the former the work occasionally varies from nil to full power, while that of an engine driving a rail mill changes from nil to practically full power while rolling a single rail twelve times, which requires about one minute.

Let us for a moment think of a flywheel 26 feet in diameter, 16-inch rim, making from 70 to 80 revolutions per minute, rolling a steel bloom into a rail. When about one-half done—say, about 16 feet in length—instead of the piece going straight out through the rolls on the guides, as intended, it sometimes wedges in the groove, and in less than a second of time it has completely encircled the roll, and forms what in rolling-mill phrase is called

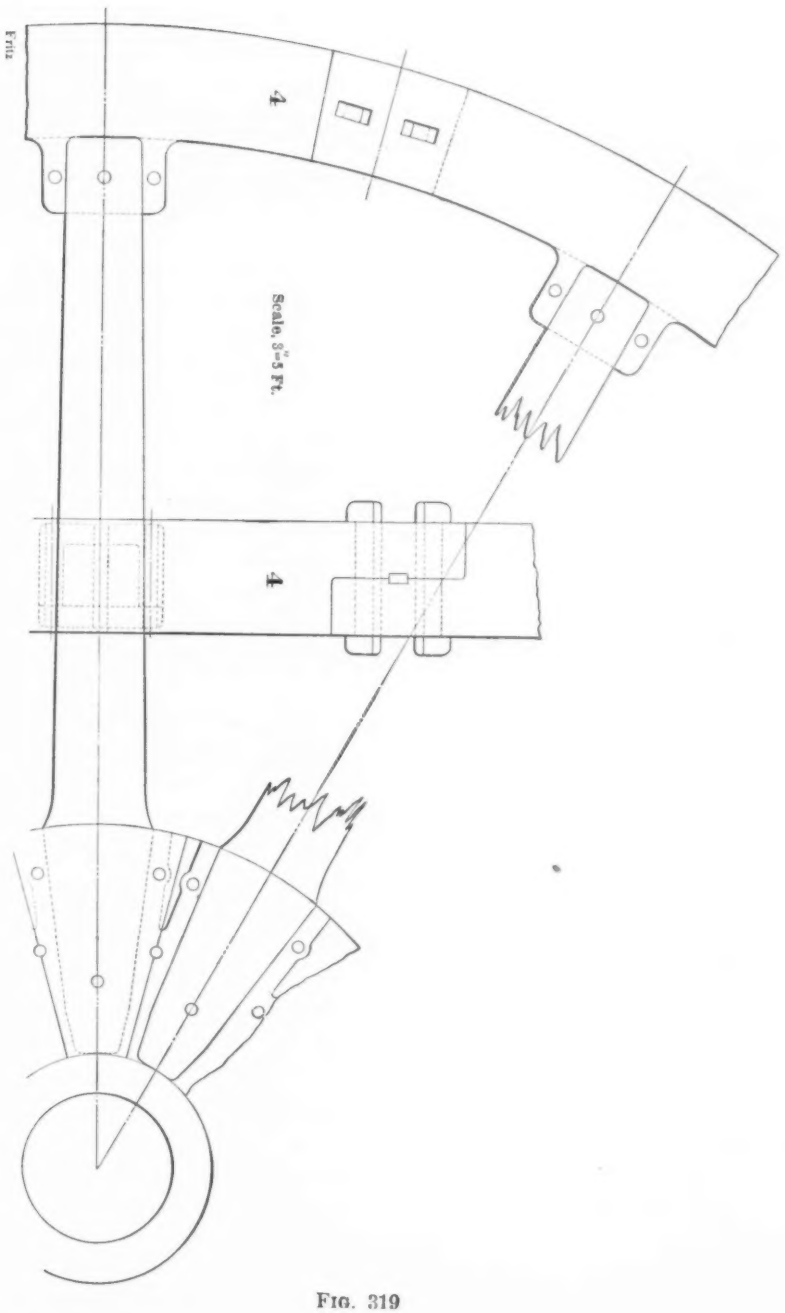
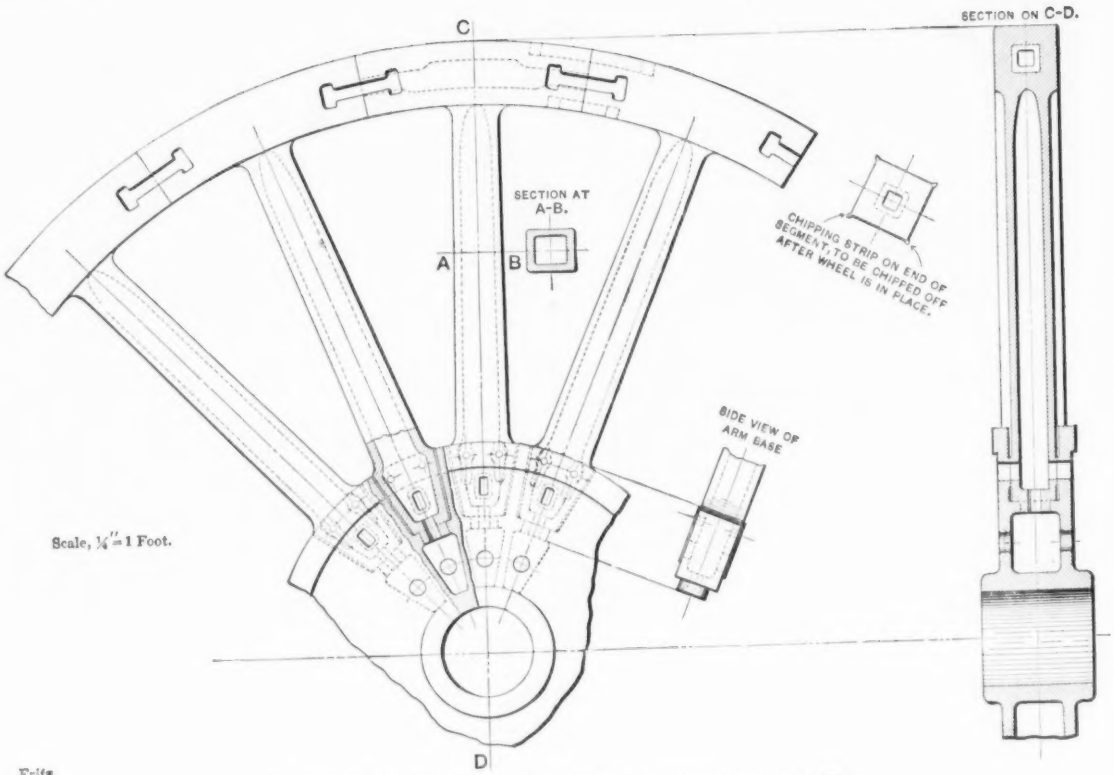


FIG. 319



28" RAIL MILL & 33" BLOOMING MILL ENGINE, FLY-WHEEL.

FIG. 321.

a collar, and continues to wind around the roll until something must happen, either the breaking of a roll, a spindle, or coupling, or the engine must be brought to a standstill in an almost incredible short space of time. The torsion on the spindles and couplings is so great that it causes the wheel to rebound. To calculate the strain the wheel is subjected to under these conditions will require some one better posted in mathematics than I am.

The fact that the introduction of steel in various forms required larger rolls, heavier wheels, and larger in diameter, which made it impracticable to use a solid-rimmed wheel, led to the adoption of plan of wheel, Fig. 321; the tracing makes it so plain that it wants but little explanation. The segment is cast hollow, and also the arms, which are made at the ends to compare in thickness to the segment, so as to relieve them of strains which might occur if the segments were cast solid. The holes in the segments are small at the ends, so as to make up for the metal taken out for the tees. The links, or tees, are different lengths, so that the strain on the segments will not come all at one place, and by using oil-tempered steel in the links, or double tees, the rim will be practically as strong at the joints as it is elsewhere.

In fitting up the wheel, the centre is bored out, and the ends of the hub are faced off. It then is taken to the planer, and a centre-guide plate for the hub is made, and bolted to the planer table. Then a boring bar is arranged to bore out the hole and face it off to a given distance from the centre for the shoulder of the arm to rest on. There are small chipping pieces on both inner sides of the receptacle for the arm. They are planed off at the same setting, to the proper and uniform width, and true to the centre. This finishes the centre.

Next the arm goes on the planer and is set true to the average centre, and with a socket on the end of the boring bar with inside cutters the end of the arm is turned off, and the shoulder is faced off to the proper length for the arm to rest on; so at this setting the chipping piece on the upper side of arm, as it lies on the planer, corresponding with the same in the centre, is planed off, and the average centre of the segment is also carefully taken and correctly worked.

After the arms are thus far finished, they are placed on the planer again, the other side up, and a socket is made to fit over the

end of the arm which goes into the socket and against the shoulder. On the opposite end of the socket is a hole bored, say, 1½ inches or 2 inches in diameter. The length of arm must be, from the centre of the hole to the bottom of socket which arm rests on, the same as the distance from the face upon which the arm rests to the centre of the hole in shaft.

Next secure a pin on the planer bed to correspond in size with the hole in the end of socket piece. Then put the socket on the end of the arm and on the pin; then bolt the arm down true on the side that is planed, and finish to the proper size. When this is done, find the exact length of the segment and set it in line with the centre, and you have the exact angle, and when all is done correctly the wheel will fit together without a chisel or a file being used on it, and will run true, without turning, minus the irregularities in the castings.

You will notice there are no abrupt changes in the thickness of the castings, thus avoiding as much as possible the liability of the strains. The joints in the segments are generally the weak part of the wheel. In this respect a wheel made from this drawing will be stronger than the original, as the tees in the segments in the wheel now in use were made of wrought iron, and will probably not stand over 50,000 pounds tensile per square inch, while in this wheel, if I was to build another, I would use steel that would stand at least 85,000 pounds tensile. This would make the joints practically as strong as the segments.

It will be noticed on tracing there is a space in the centre of about ¼ inch in front and rear side of each arm. This is filled with oakum and driven hard, after the wheel is finished and in place, to keep the arm from yielding in the direction of the strain, and at the same time greatly lessens the work of fitting up the wheel. The 1¾-inch round holes shown on tracing, through the centre and arm, are reamed out, and steel pins made and turned so that they will drive in snugly.

The bolt shown in segment in Fig. 318 does not belong to this wheel, but was simply put in there to show the plan that, in many cases, was adopted to make the wheels more secure, and is used to some extent at this time.

DISCUSSION.

Prof. C. V. Kerr.—I owe Mr. Fritz a debt of gratitude for bringing out one point in his paper, and that is in connection with Fig. 320. As a part of my regular work in steam-engine design, I paid some attention to the question of the stresses in flywheel rims and arms, and I have been for some time harboring a proposition in regard to the construction of flywheels which was a little startling to me at first, and I hesitated to say anything about it; but Mr. Fritz having made the statement that he does in regard to construction in connection with Fig. 320, I see that my idea is correct. When such a flywheel as this is running at any given velocity, the centrifugal force of the rim puts a uniform radial load on the rim. Now if, as is usually the case, the rim is made in one piece or bolted rigidly together, and the arms are fixed solidly in the hub and rigidly at the rim, it constitutes of that flywheel rim essentially a continuous girder in which the points of support are at the ends of the arms. The effect of this centrifugal force acting radially is to bend that rim outward, so that, starting at the end of the arm, you have a reversed curve. That fact puts a bending moment in the rim at the end of the arm, such as if this were carried to the point of rupture it would begin to break, and it also puts another maximum moment at the middle of the segment tending to break it on the outside first. Now, the moment at this point, the end of the arm, is approximately double what it is at the middle of the segment. Consequently, the greater strain, if it is a solid rim, is at this point. If, however, as Professor Benjamin's experiments very plainly showed, the joint in the middle of the rim tends to open out, it would take this V shape. That relieves for the moment the stress at the middle of the rim and puts it at the end of the arm, so that it increases the stress at that point. So far as the arm is concerned, you see it is the function of the arm to put this bending stress into the rim. Consequently, we have to go for safety to one of two extremes. If, as stated in the discussion of Professor Benjamin's paper, the arm strengthens the rim, then the more the better; and that is true in a sense, because the moments at the outside and the inside of the rim depend on a square of the length of segment. Consequently, the more arms you put in, the smaller will be these moments,

and that extreme will lead to putting in arms enough to make a solid disk—which of course we do not want on account of the weight. On the other hand, if we put in arms not fixed at the rim, then we relieve this bending tension entirely, and it becomes simply a question of hoop tension. Now, that is the conclusion to which I came in considering the stresses in the flywheel rim. Fig. 320 of Mr. Fritz's paper shows exactly that condition. It is true that a little oakum was driven in between the arm and the rim lug in order to keep the arm in place; but we cannot for a moment conclude that it prevented this rim from expanding as it wanted to do. In view of the fact that we must have arms in order to drive the rim, then it seems to me that the best construction would be to leave the bolts loose at the rim. That is, if you turn to Fig. 318 in Mr. Fritz's paper, then the bending stress in the rim would be entirely relieved, simply by backing off the nuts on those bolts which hold the arm to the rim. It will occur to you, probably, as it did to me, What is to be the effect on the running of that wheel? Won't it get to wabbling, especially under belt pull and under the effect of gravity? If you take a 24-foot wheel, which probably is about the size of your rolling-mill wheels, and if that runs at, say, 72 revolutions a minute, that will give it a rim velocity of between 90 and 100 feet per second. That would give a centrifugal tension of probably 800 pounds only in the rim. Allowing, say, 12,000,000 as the coefficient of elasticity for cast iron, that would give an elongation of the entire rim of about $\frac{6}{1000}$ of an inch, and if you put that upon one arm it comes down to $\frac{1}{1000}$ of an inch. Consequently, the amount of swelling of the rim due to the centrifugal force is not enough to occasion any trouble in regard to the driving. You will notice that so far as the effect of gravity is concerned in a six-arm wheel, it will be supported by the bolts or pins at the ends of the horizontal diameter, and to support the belt pull there would be the bolts at the end of the vertical diameter, so that so far as that goes there would be no trouble.

I consider this a really important point. I have supervision of three engines, one of which, a high speed engine, has a bending tension in the rim at its ordinary speed of about 4,800 pounds, and a centrifugal tension of about 350. Another one, a Corliss engine, with 16-foot flywheel, has a tension in the rim of about 4,200, and a centrifugal tension of

about 320. Another wheel, 6 feet in diameter, has about the same centrifugal tension and 7,800 pounds bending stress. Now, flywheels have usually been designed simply on the basis of this centrifugal tension, and I have just mentioned a case in which the ratio of the centrifugal tension to the bending tension is 22 to 1. For similar reasons a good many of the wheels have gone to pieces. I would like to refer you to an illustration in *Power* for November, 1897, in which a 13 foot flywheel ran away and burst. The wheel was put together again after the accident, and it showed that the breaks were exactly at the poin's indicated by analysis. So far as the arms of the flywheel are concerned, leaving them loose at the rim would be somewhat to the disadvantage of the arm, for the reason that ordinarily the arm under the action of inertia or belt pull tends to deflect into a curve which is more or less like the middle half of the letter S, with a maximum stress at rim and hub. If the arm were left free that would put a maximum stress at the hub, which would be approximately one-half greater than it is with arms fixed; but we will relieve the tension due to centrifugal force of the rim segment supported, so that with the same material we can make an arm fully as strong.

There is another article to which I would like to refer—*Power* for January, 1894—where a wheel suddenly stopped, broke all the arms at hub and rim, and the rim dropped down into the pit. That was to me at least a demonstration which was not to be gainsaid in regard to the stresses in flywheel arms. Moreover, these stresses, which in fact occur periodically during constant speed and on opposite sides of the arm, are reversed during each stroke.

Mr. William Kent.—I have recently seen a flywheel which violates Mr. Stanwood's principles in having the joint half way between the arms. The rim was not very thick, not over 3 inches, and the width was probably 15 inches. It went to pieces by breaking through the flange. The section of the break showed a lot of dirt, blow-holes, and cinders. It seemed as if all the rubbish out of the end of the ladle came right to the most dangerous point of the flywheel, and the discovery was made that inside of this fractured piece was a lot of pieces of babbitt metal, or some other metal, which had been put in there to hide the defect, but it had not completely filled it. Nobody was hurt by the accident, but if there had been somebody hurt

or killed there might have been an action for damages or manslaughter that would have led to the punishment of the criminal in that foundry who put babbitt metal in a flaw in the most dangerous part of an engine.

Mr. Oberlin Smith.—It seems to me that it is a disgrace to this age of mechanical engineering to have so many flywheels bursting all over the country. We talk about it and propose a hundred different ways of keeping them together, yet we do not always attain our end. The time has come when something should be done. Undoubtedly a flywheel rim should be of forged metal if possible, though it does not seem practicable in all cases. It is rather painful to see the manœuvring there is with cast-iron wheels to keep the rim from breaking the spokes, and the spokes from breaking the rim. Of course when we cast the wheel all in one piece we have on the rim the bending action of the spokes, and if we fasten the spokes separately to the rims in small wheels we fail to get economy in cost. My experience with small flywheels up to 4 or 5 feet in diameter has been that, if they are properly cast, so as to relieve the initial tensile strain on the spokes, there is not much trouble with them, but even then the spokes will crack sometimes unless great care is taken in casting. The best way seems to be to take the sand off the middle of the wheel first, not enough to stretch the spokes and crack them as they shrink, but just enough so they can shrink and be put under some tension. Then afterwards as the rim cools it contracts upon them and relieves such tension, putting the wheel approximately in a state of equilibrium.

Leaving these small wheels and speaking of the larger ones, Mr. Fritz's general idea of a complete rim cast in one piece, and spokes cast separately, is undoubtedly good. His spokes, which are simply caulked into a recess in the rim with oakum, practically form a loose joint, and I here suggest as a substitute (for cases where it would be suitable) a simple pin-joint, thus, as in Fig. 322. In this case we would have a rim made up in one piece, or, if it was too large, made up in several pieces, rigidly fastened together so as to practically form one piece, and that rim would take care of itself if strong enough. It seems absurd that we should put a lot of artificial bending stresses into a rim in addition to the normal tensile stress which it must have.

With my pivoted joint (which probably is not new, although I have never seen it used) we have a neat job, and the spokes will take a part of the outward pull of the rim. One objection to making the spokes thus loose at the outer end is that we are using them as one-ended beams, which require considerable lateral strength in comparison with ordinary spokes, acting as beams of double support. But we simply need to make the spokes of proper proportion, and wider perhaps than usual, so as to get strong beams. If we do the other way, fastening the spokes rigidly to the rim, we are getting a lighter, cheaper, double-ended beam, but are putting a lot of bending stresses into our rim, which has all it can do (and sometimes more) to

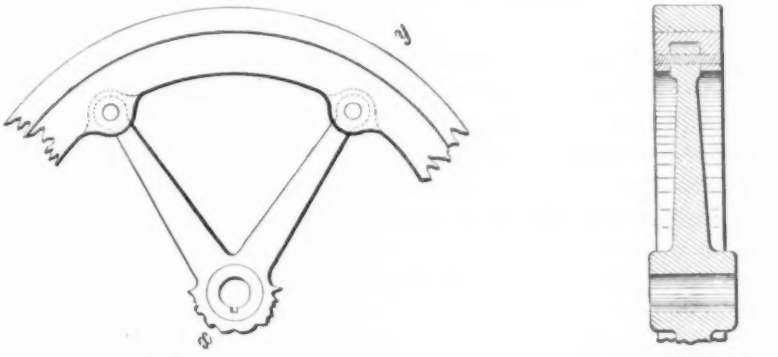


FIG. 322.

withstand the centrifugal force alone. It therefore seems to me that the coming large flywheel will have, wherever possible, a separate rim with spokes put loosely into it in the way that has been suggested by Mr. Fritz, or perhaps with simply a pin-joint. The spider would thus be a thing of itself, entirely relieved of initial strains, as would also the rim. A still better rim, of course the ideal one, would be a forging or a built up structure of forged metal. It would not be a matter of great difficulty with moderate sizes to cast a rim with the proper spoke lugs, and shrink upon it a forged tire, or to wind it with wire, preferably of square cross-section.

Mr. Gus. C. Henning.—I have had occasion at a previous meeting to point out why Mr. Fritz's wheels ran so well, and in fact it was suggested that none ever failed—"that they were all

right." The point in the design of flywheels to which Mr. Fritz has given more attention than anyone else is this, that he eliminates all the weak material in every part of the wheel, at the same time eliminating the possibility of dangerous shrinkage strains in the corners, and also making provision for uniform shrinkage of the different parts of the wheel, so that no matter what the shrinkage is it is harmless. This can be done by casting the arms in the manner shown. The material in these sections, as shown in the arms and in the rims, is always in the best shape. The material at the centre of solid rims, hubs, and arms is always in the poorest shape. In fact, it is frequently hollow or segregated, and we rarely ever know the actual strength for this mass of material. The material in the ribs joining the arms and rim is more rigid than the other, and its rigidity, as I pointed out last fall, makes the wheels good and strong. Keying the wheel together where the greatest bending moments occur, as shown by the sketch, and providing the strongest material at this place, as Professor Kerr showed, leaves the material in the best condition, and united in a thorough manner where it would otherwise be weakest. I think the advantage of using cast iron in the shape in which we know it best is one of the essential points in the wheel. A hollow rim would certainly be much better in every respect, properly cast, than a solid rim, because the material would be in a known condition which we know to be reliable. Of course it is well known that this oakum packing makes an elastic connection, and when the inertia due to the reversal of the roll train or a sudden stoppage acts through the rim of the wheel and the arms there is always a certain amount of compressibility of material at the hub which prevents all of the load coming entirely on the arms suddenly. Of course, this shape is more rigid than the same amount of solid material would be; that is, the same weight of material in this rim makes it very much stiffer than it would be if it were solid. Taking all these matters into consideration, I think it explains why these wheels have never failed.

Mr. Jas. McBride.—Mr. Fritz at the last meeting promised to show us a flywheel of his design that he had built and run for years, and which had never broken. There we have it, and I think that he has in the design of that wheel eliminated every weakness. It has been sufficiently strong for thirty years to do the work for which it was designed.

Mr. Kerr.—If we are to enumerate the names of those who have investigated flywheel stresses, then in addition to that of Mr. Stanwood we ought at least to mention that of Professor Lanza, and, I believe of an earlier date, that of Professor Unwin. The fact is that the speeding up of flywheels, especially those on Corliss engines, due to the demands of electric-lighting service, has been largely responsible for the large number of flywheel accidents. The factor of safety in those wheels was already small when first designed, and the speeding up simply caused them to pass their point of safety. Now, as Mr. Fritz has already used this method of construction which I have proposed, I would like to see the type of band flywheel tried by some one. I think it is safe, in fact the safest form of flywheel that can be designed. Numerous sections of flywheel rims, somewhat fantastic, have been proposed; but I think the safest section is simply the plain rectangular section with the arms free—that is, that the rim expand under its own tension. The references which I made to *Power* will, I think, prove conclusively to those who will investigate the circumstances the actual facts that the maximum stresses in wheels do come as has been indicated here.

In justice to myself I should add that my note books show original and complete analysis of the stresses in flywheel rims and arms of a date at which I did not have the benefit of the work done by Professors Stanwood and Lanza. There are no doubt others who have investigated these stresses independently.

Mr. Wilfred Lewis.—I have just one suggestion to make. It has been pointed out that the danger to flywheels is due chiefly to bending strains, and that the construction shown in Fig. 5 tends to relieve the bending strains through the elasticity of the oakum packed in the joint. Possibly a looseness in the joint would be favorable to the distribution of the strains in the rim and arms, but it could not be advocated as good practice. Now, it seems to me that the proper way to relieve the bending strains while running is to set up bending strains when the wheel is standing, and that the construction shown in Fig. 321 lends itself particularly to the accomplishment of that end. If the arms are put in compression when the wheel is at rest by wedges driven in on top of the arm, instead of leaving it loose, as suggested, then the rim would be under bending strain when standing, and when running at the proper speed it would be

entirely relieved of bending strain, and subjected only to hoop tension, which is more easily taken care of.

Mr. James B. Stanwood.—I have recently had occasion to investigate the strain thrown on the rim of a wide band wheel by the usual form of lug located midway between the arms. I find that such a lug will often double the strain on the rim, besides weakening the rim itself.

It has been my pleasure to visit Professor Benjamin at Cleveland and to have inspected some of the burst wheels upon which he has experimented. At the last meeting of the Society I called attention to the fact that if the lug was located near the arm, say about one-fifth to one-fourth of the entire span of one arm, then I believed that wheels so constructed would be stronger. I hope that Professor Benjamin will be able to give us some experimental demonstration on this case.

In regard to wheels in general, there is less danger from the large square rim wheels which are stiff in the rim in relation to the strength of arms, and which are usually constructed under the eye of a competent engineer, than the "commercial" band flywheels and pulleys, which are designed and used in an indiscriminate manner. If some simple standard proportions and design, based both on practice and theory, could be evolved for such wheels there would be less loss of life and property than now.

Prof. Forrest R. Jones.—During the last year I was investigating different flywheels in order to see what might be made by our representative builders. I found three that seemed to be worthy of particular attention. One was made by the E. P. Allis Company, of Milwaukee. It had tangent arms placed somewhat as in a bicycle wheel, the arms being made of rectangular-section bar iron, however, and so attached to the rim that the side-bending action of one was counteracted by that of another. This particular flywheel was designed for a mine-hoisting engine where the variation of load was very great.

Another, by a different concern, was a band wheel of a form which has not been mentioned here to-day, in which there is a very broad and comparatively thin rim. The manufacturers of this wheel had noticed that fractures very often occur in such wheels in the arms, so their idea was to design an arm which would not be apt to break. They made the arm hollow, larger at the hub, both in diameter and thickness, than at the rim. Through the centre of the arm they put a rather large tension

rod of steel, with either a key or nut on one end for tightening it, so as to put the cast-iron part of the arm in compression by putting the steel rod in it in tension, thereby relieving the cast iron of the tension partly or wholly.

Another form, made by two or three manufacturers, I believe, is that in which the arm is of an I-beam cross-section, the web being placed in the plane of the wheel, this wheel being particularly adapted to resist strain coming from variations of speed due to sudden loading of the engine.

It was stated a few minutes ago that the wheel sketched on the board is the safest form of flywheel that can be designed. While this may be true of cast-iron flywheels, there are designs in other materials which will unquestionably stand higher speeds and greater variations of speed. Notable among these are the following two, one made of plate steel and the other wound with wire.

The steel-plate wheel has a web made of a number of segmental plates placed so as to break joints, and held together by rivets passing through from side to side. The rim is built up of several pieces of plate metal cut so as to form a ring when placed together. These sections are placed together so as to break joints, and are riveted through from side to side. Steel plate 1 inch thick is used for the web, and $1\frac{3}{8}$ and $1\frac{1}{2}$ inches thick for the rim. The hub is of cast iron, 72 inches in diameter, made of two disks. It can be seen that the pulley has what might be called a built-up web. The pulley is 18 feet in diameter and has a rim $15\frac{1}{2} \times 16$ inches. The web is made of two thicknesses of 1-inch plate. The factor of safety is about forty.

The wire-wound flywheel, having a rim composed of wire wound circumferentially under tension around the web, was constructed for use in a rolling mill at Landore, Wales, using the Mannesmann process of rolling tubing from the solid bar. In this class of rolling a very great amount of power must be supplied to the machinery in a short time. The strain upon the flywheel is accordingly very great, on account of the sudden reduction of speed which it must undergo in order to deliver its energy, as well as on account of the high speed at which it runs in order to have sufficient energy stored in it.

The construction adopted is the strongest and safest that can be devised. Two steel disks, 20 feet in diameter, are bolted to a cast-iron hub. The outer edges of the disks form a groove

into which wire is wound to form the heavy rim. The groove is filled with 70 tons of No. 5 steel wire wound on under a tension of 50 pounds. The wheel is run at 240 revolutions per minute, which corresponds to a circumferential velocity of 15,080 feet, or 2.85 miles per minute.

Mr. Boyer.—I would like to ask Mr. Fritz if he ever knew of a flywheel of that character going to pieces.

Mr. Fritz.—I never did.

Mr. Boyer.—I recall by my earliest recollection flywheels built of that design, and I have no recollection of an accident. I will say to all these fellows that have built wheels that are breaking to go and design likewise.

Mr. Oberlin Smith.—I am glad to hear mentioned the wire-wound wheel, and there is no question in my mind but that the truly good flywheels of the future must be either banded with hoops or wire-wound. We should, in making these rims, use the same amount of painstaking care to resist the maximum bursting stresses as we do with our cannon—begging pardon of the naval gentlemen present, I should say, rather, big guns. In regard to wire spokes, or other suspension spokes of any kind, I think there is no question that they can be made to do good work for a rough wheel; yet an objection to them would seem to be that their undue elasticity would lead to the rim not always running as true as is required for belting purposes, especially as the belt gives a heavy lateral pull. It would therefore seem better to use a good, stiff cast-iron spider, preferably, perhaps, with the spokes rather large and cast hollow, of elliptical section. Such a construction, having them pin-jointed to a steel-hooped rim, would, I am inclined to think, make an ideal wheel.

Mr. J. A. Brashear.—I desire to bear testimony to the efficiency of the wheel designed by Mr. Fritz. During my rolling-mill experience of over twenty years I have constructed a number of flywheels of this design, and in but one case has one of these wheels been destroyed; that one under quite anomalous conditions—namely, by the burning of the rolling mill. The heat was so great that the expansion of the rim pulled the flange sockets away from the hub, but by making a new hub the wheel was reconstructed and is still running. I think this construction of a large flywheel is practically perfect, provided the castings are carefully made and allowed to cool slowly so

that there shall be a minimum of molecular strain in the metal, particularly where there is a change of direction in the shape of the casting.

I have seen several flywheels go to pieces, and I believe in almost every case the breakage was due to molecular strain in the material.

Mr. Smith refers to a solid circular rim cast flywheel. I very well remember an experience with such a wheel. The owner of a rolling mill in which I was employed determined that he would cast a solid-rimmed wheel of about 26 feet diameter, and to weigh when finished approximately 40 tons. The rim was of an oval section about 14 by 12 inches in the major and minor axes. Steel arms 4 inches in diameter were cast with the rim, and the centre or hub of the wheel was to be cast after the rim had become almost cold. The mould was made close to the flywheel pit and the metal run in an iron-lined trough across a roadway from the old foundry of Knapp & Wade, celebrated during the war for casting the Rodman guns. The casting was successfully made and allowed to cool for nearly a week, but it cooled too rapidly, I think owing to the thin casing of sand in the mould, and it went to pieces with a shock that was felt for squares around. The rim did not break at symmetrical points, such as at the arms, where one might think rupture would take place, but between the arms, as well as near them, showing tremendous molecular strain. I am certain, from a long experience in working material, that molecular strain is a most potent factor which should be studied very carefully by the mechanical engineer.

Mr. Jno. T. Hawkins.—I would like to say one word, which I am glad to say will be in corroboration of Mr. Fritz and his flywheel. In the experiments of Professor Benjamin, described at the last meeting, I believe his flywheels were all ruptured by centrifugal force only. I think there is one feature in flywheels lost sight of in this discussion, and that is that there are two distinct varieties of strain brought on a flywheel. The fact that it is properly proportioned and made to resist one variety of strain is no indication that it is able to resist the other. The large majority of flywheels which do tear themselves to pieces do so on account of high peripheral velocity and the resulting centrifugal stress, particularly where the results are disastrous to the surroundings from the rim flying to pieces.

Now, flywheels are also constructed which do not run and do not require to run at very high peripheral speed, but are constructed to store up energy during a portion of the revolution and give it out at other portions, which is a somewhat distinct thing from the other and requires altogether different proportions and is a different construction really. I quite agree with Mr. Smith that it is somewhat of a reproach to the engineers that there are so many flywheels bursting throughout the country. Where a flywheel is intended to withstand high rotation, without any severe efforts to arrest its rotation at different points of a revolution, it requires a very different structure from that now illustrated by Mr. Fritz. This is a typical case of rotative effort that is to be stored up during a portion of the revolution, or perhaps during two or three revolutions, and suddenly given out, and I apprehend myself that the success of this particular wheel in the rolling mills is partly due to the fact that its connection with the hub is made somewhat elastic. Rolling-mill stresses more or less suddenly resist the rotation of the rolls and are somewhat of the nature of a shock, and when we get down to absolute shock we are liable to be shocked ourselves at the result. We do not know just exactly how to deal with it, and the least elasticity given to a structure of that kind really takes the maximum danger away.

Mr. Fritz's flywheel is clearly and properly proportioned to resist that kind of stress which tends not to produce peripheral stress in the rim so much as to produce rupture at the juncture of the arms with the rim and the hub; it does not appear, therefore, that the analysis of the stresses brought to rupture the rim, given by a previous speaker, applies to the type of flywheel described by Mr. Fritz. Flywheels required upon punching, shearing, and drawing presses are types of what Mr. Fritz has described, and all are quite distinct, for instance, from steam engine flywheels.

Mr. Oberlin Smith.—We cannot always know the molecular condition of the iron in the interior of a flywheel rim, nor how much it is affected by shrinkage strains. It is all right to have the best quality of iron, but we must remember that we have the ever present blow-hole; and that sometimes there are hidden blow-holes, so that we never know what the real cross-section is at every point of the rim. Cast-iron rims subject to centrifugal forces must be, after all, nothing but a lottery and

a venture—and this fact *cannot be impressed too strongly* upon their makers and users.

Mr. Fritz.—I propose to occupy your time but a moment. In the first place I am not a talker. Prof. Kerr spoke about leaving the arms loose. My experience all my life has been to make everything fast. I do not think it is possible to make them loose without getting into trouble. I undertake to take care of the strength which he spoke about, in this way. In these rims there is no possibility of their getting out of a true circle unless the engine runs away, or some remarkable accident happens to it. They are made heavy, and being cast hollow add depth to the rim, and being securely fastened at the ends they form a continuous girder. In regard to the tendency of the wheel to go to pieces, I have said in my paper that I know of no class of work that is so severe on a flywheel as that of the rolling mill, as no work could well be more irregular.

Another thing Professor Kerr has mentioned, the importance of getting the right kind of metal, and getting it so distributed as to avoid, as far as possible, all casting strains. This is all-important, and is one of the great advantages of the hollow rim, and had much to do with its adoption.

A word about cooling. My experience has been, when a casting is made, cover it well, letting no part be exposed to the air, so that it will cool uniformly. Another thing about the metal. Get it as strong as you can, avoiding all tendency to check. We aimed to get 22,000 pounds tensile. Probably it is not as high as that, and will range between 20,000 and 22,000 per square inch.

In regard to castings, I have reference to castings that are fairly well proportioned in thickness.

I can only say, in regard to this form of wheel, there are fifteen or twenty running, varying from 12 to 30 feet in diameter. There has never been a particle of trouble, never a tee loose or anything else, to the best of my knowledge. One word about band wheels. I was asked to make a drawing of one, and Fig. 235 shows such a one as I would use for a rolling mill should I be in want of such a wheel.

Mr. F. A. Halsey.—So much has been said about the bending strains in flywheel rims that it should be pointed out that Mr. Fritz's design does much to minimize those strains by providing

more arms than usual—sixteen in the wheel shown. This is an improvement, not only because of the increased strength due to the arms, but because the increased number of arms reduces the length of the individual segments of the rim. If the depth of the rim section, the fact that the rim is a box girder, and the small amount of overhang of the segment are considered, it will be seen that the bending strain on its rim must be very small.

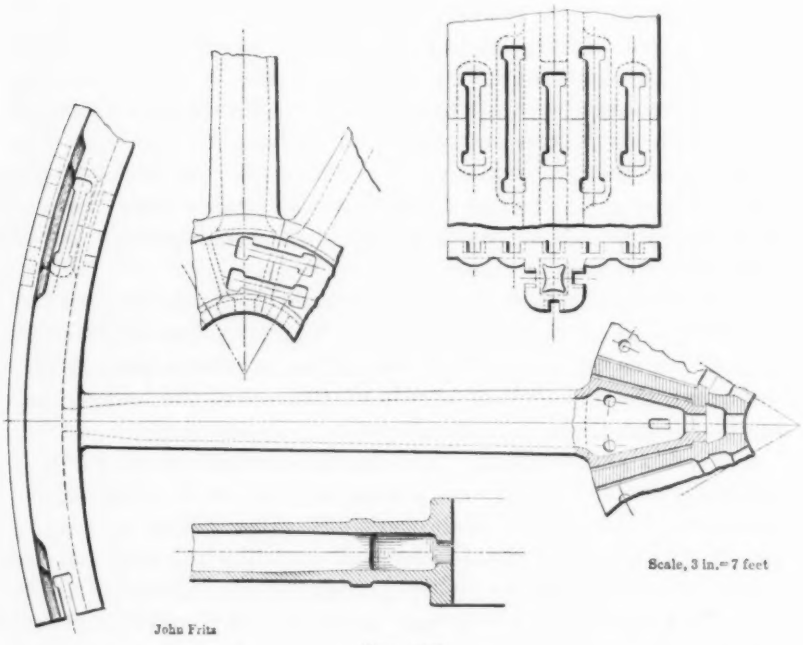


FIG. 235.

It is more than probable that in Mr. Fritz's wheel the arms are a source of strength and not weakness.

Mr. Fritz.—In that wheel I tried to get in as large a number of arms as possible. That wheel has sixteen arms—10 inches wide where it enters the hub at centre, and 13 inches at top of billet, and they are 12 inches the other way, so that the wheel is perfectly stiff in both directions.

Mr. Comly.—Mr. Fritz's idea of having a good many arms is certainly in the right direction. It seems to me that we ought to go further and build them up out of steel plates in which the rivets or bolts would be made 6 inches or so apart around the

rim; that would be a step still further. Another point about flywheels and pulleys is the question of the dust collecting on the inside of the rim. A flywheel of this sort is not so much trouble, but in some classes of manufactures, a wheel on which there is a belt is a very serious trouble, especially where the wheels run night and day, and from one year's end to the other. It seems to me that a wheel built on a girder principle of steel plates is a step in the right direction. Some of them are now in use and giving very satisfactory results. I hope that steel pulleys for belts will receive careful consideration by our pulley manufacturers, as well as heavy steel flywheels.

Mr. Fred. A. Waldron.—We are all ready and willing to acknowledge that Mr. Fritz's paper is concise and to the point, and we respect the opinion of his associates, and defer in all questions regarding cast-iron flywheels in favor of Mr. Fritz's method of construction. Experience for the past twenty-five years has shown them to be safe, and in all probability they are good for twenty-five years more.

Conditions, however, are constantly changing. Electricity has entered the mechanical field in the last decade, and with its rapidly advancing strides demands more perfect steam engines. Closer competition demands the greatest economy in floor space and steam consumption. To meet these demands the engineer is in the field with high piston and rotary speeds, which even to the present day have not in all probability reached their limit. To meet these conditions, commercially and theoretically, engineers all naturally turn to a stronger and more certain metal than cast iron.

Mr. Fritz and his associates in the rolling-mill business have forced this condition upon us in giving us wrought iron and steel which are from four to six times as strong as cast iron, and of an infinitely higher elastic limit, together with an absolute knowledge of their physical properties. This material can be bought in open market, rolled to special shapes, in large quantities, from ½ to 1½ cent per pound cheaper than cast iron.

With this condition confronting us, I predict the flywheel of the future, whether of the balance-wheel or driving type, will be of wrought iron or steel construction, built up from the hub with steel plates properly braced as spokes, and a built-up rim, made by dovetailing and riveting or bolting the segmental plates together.

In the case of driving wheels used as flywheels, a modified channel or angle section, rolled to suit, shall constitute the rim, with a single steel-plate web or arm in the centre for narrow rims, and a plate attached to each side of the rim and braces laterally for wider rims.

Wrought-steel pulleys are now obtainable in open market up to 24 inches in diameter. These can be bought as cheaply as cast-iron pulleys of corresponding size, if not cheaper. They are lighter than cast iron, and, if properly made, are far superior.

Wrought steel stamped accurately can be obtained in various forms from a small-sized padlock up to the larger portions of boilers of every description, and the end is not yet.

DCCCXXIV.*

A NEW SYSTEM OF VALVES FOR STEAM ENGINES, AIR ENGINES, AND COMPRESSORS.

BY FRED. W. GORDON, PHILADELPHIA, PA.

(Member of the Society.)

In steam or air motors there are four valves in each head, two for the inlet and two for the exhaust, each pair being an exact duplicate of the other. A valve really consists of a pair of valves, of the same diameter, connected by links to a rock arm, at an equal distance from the centre of the rock shaft, one valve moving it into the cylinder, the other away from it to open their ports. The valves are in equilibrium as to current and pressure. If we are referring to the head end of the cylinder, and the piston is at the dead point at that end, then the ends of these valves may be flush with the inside of the cylinder head, and the only clearance will be the circular bevelling of the ends of the valves which move outward, and the ribbed chamber around the ends of the valves which move inward to open, together with whatever clearance is thought proper to avoid contact between the piston head and cylinder head. This total clearance need not exceed one-half of one per cent. for ordinary Corliss-engine proportions as to diameter and stroke. The excess of the surface exposed to the steam upon its introduction to the cylinder is not more than 9 per cent. above the area of the piston head and cylinder head. A minimum of clearance, a minimum of surface to reduce internal condensation, and maximum of portage are the claims for this system. The illustrations show the adaptability of the system for shaft-governed or automatic liberating engines. The details of these need not be dwelt upon, as they would be modified for each particular case. Neither is it necessary to call special attention to the method of packing the piston valves, as that is not essentially a

* Presented at the Washington meeting (May, 1899) of the American Society of Mechanical Engineers, and forming part of Volume XX. of the *Transactions*.

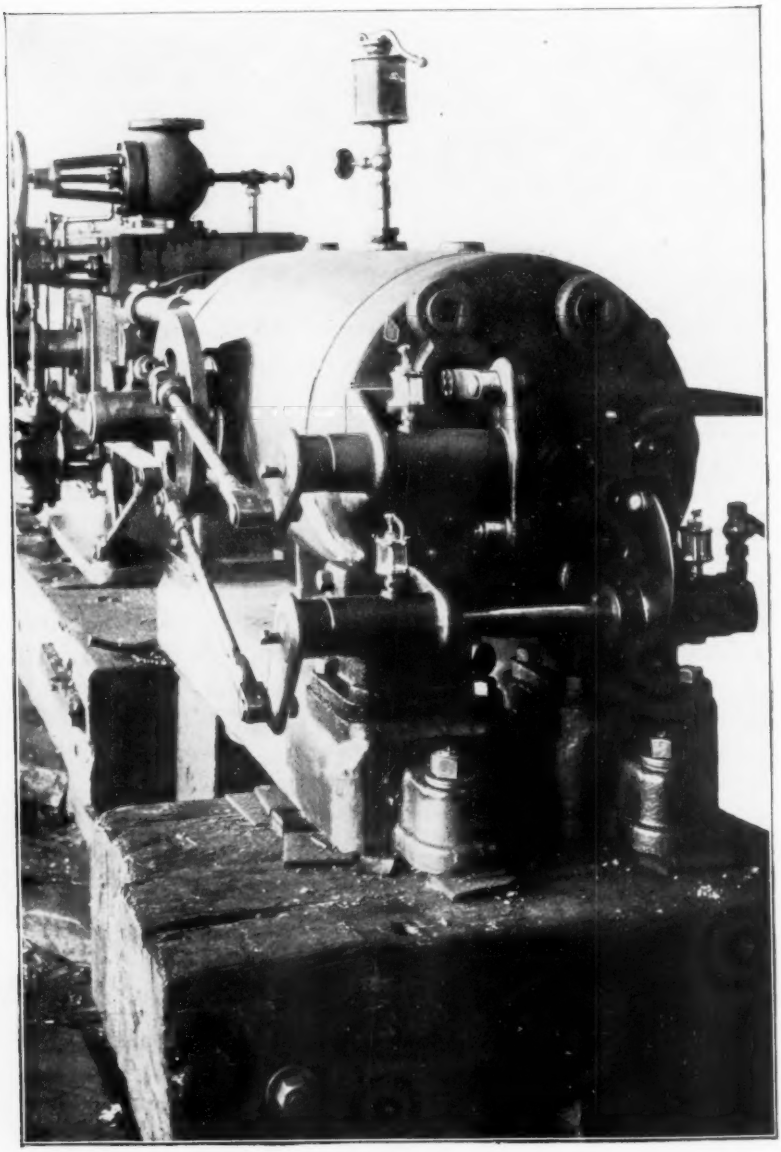


FIG. 323.

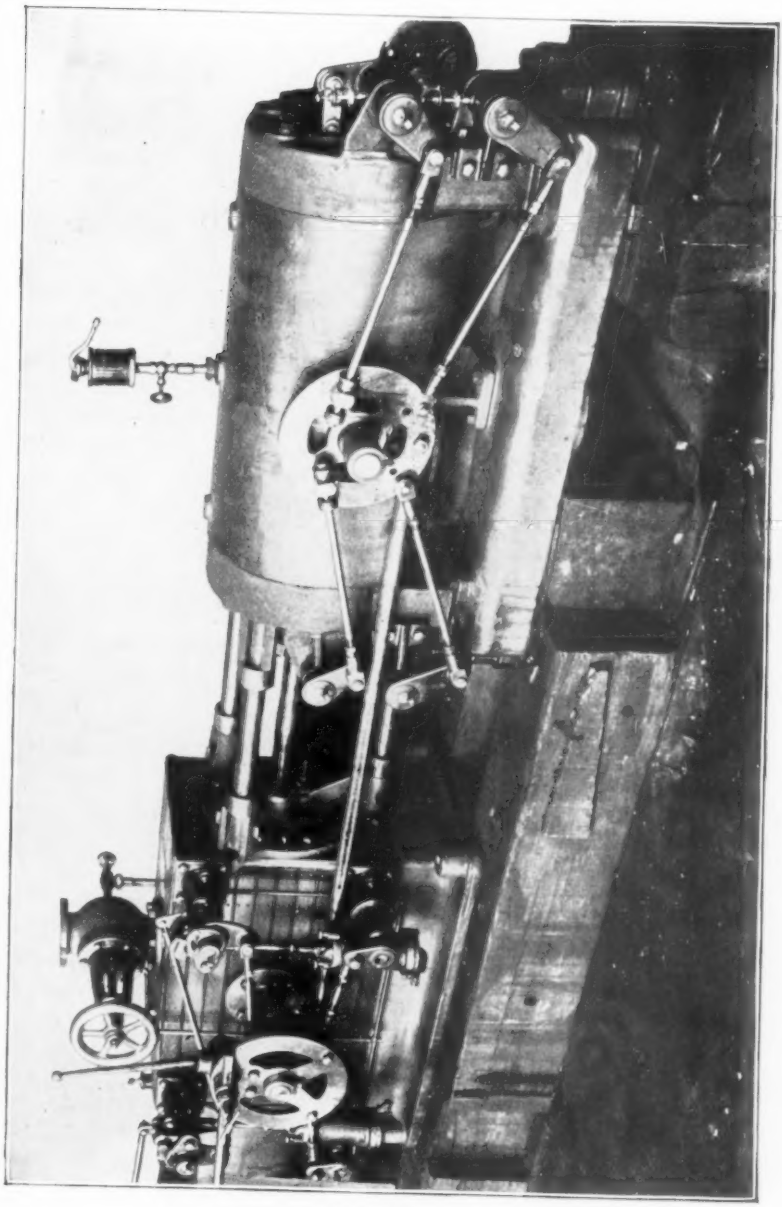


FIG. 324.

part of the system, though it is believed the double ring shown is the best—it certainly has proven to be tight.

These valves being placed in the heads of the cylinder, and the valve cylinder's bore being parallel with the bore of the main cylinder, the inner flush ends of the valves are parallel with the inside of head. The distance from the cylinder to the port is shorter than usual, reducing the resistance and change of temperature incident to long ports.

When the engine is compound the employment of the valves in the heads is conducive to direct and free passage between the cylinders, so that practically exhaust pressure in a high-pressure cylinder may be the initial pressure in a low-pressure cylinder. When the piston is at dead point, and the valves have the desired lead, and the ends of the valves nearly flush with the inside of the cylinder head, then, as the piston recedes from that end, one of the valves follows the piston and the other recedes from it. It will be seen, then, that no amount of increased travel of the valves materially increases the clearance, hence the system is especially adapted to shaft governing, where in shorter cut-offs the reduction of a small valve travel will affect the portage and cause wire-drawing.

For air compressors, which include blowing engines used at blast furnaces, gas or any other elastic fluid compressors, the inlet valves are constructed so as to be similar to one pair of a steam engine's valves. The delivery or discharge valves may be similarly constructed, or modified as referred to further on.

Fig. 325 is a sectional elevation of a pair of these valves connected by link, and acting as one valve. The outer ends of these valves at *A* are shown with metallic packing rings, to be used when the valve is employed in compound compressors or where the inlet pressure exceeds the atmospheric. This system may be used in steam engines, but we prefer that shown on Fig. 326. This drawing is made to scale, and the clearance can be accurately calculated, the valves being 3 inches in diameter and suitable for a 14-inch air compressor, and amply large for a 16-inch steam cylinder at 600 feet piston speed. The actual portage of these valves is 13 square inches.

Fig. 326 is a side elevation of a steam cylinder provided with this valve system, and arranged for releasing gear, the left hand showing the steam valves and the right-hand end the exhaust valves.

Fig. 327 shows the left-hand end of a compressor with the wrist plate operating the inlet valves. In the connection between the wrist plate and the rock arm there is lost motion, leather-faced tappets being employed. The valves being in per-

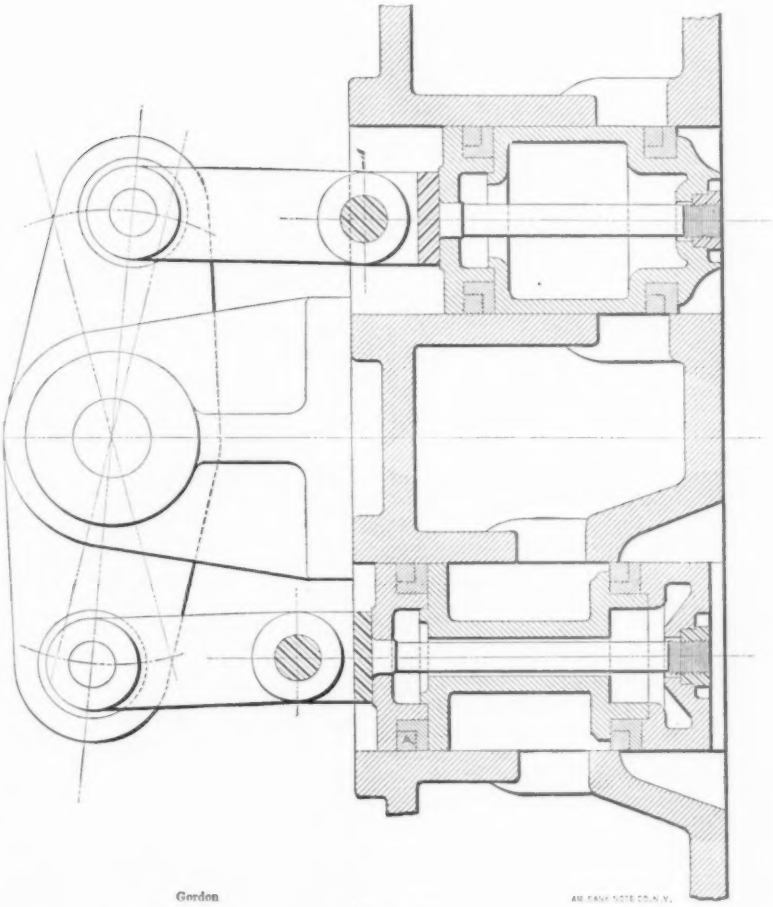


FIG. 325.

fect equilibrium, they move with such freedom that at 100 revolutions per minute these tappets make scarcely a perceptible noise.

Fig. 328 is a diagram illustrating the action of this lost motion just referred to.

Fig. 329 is a blast-furnace blowing cylinder, 42 inches in

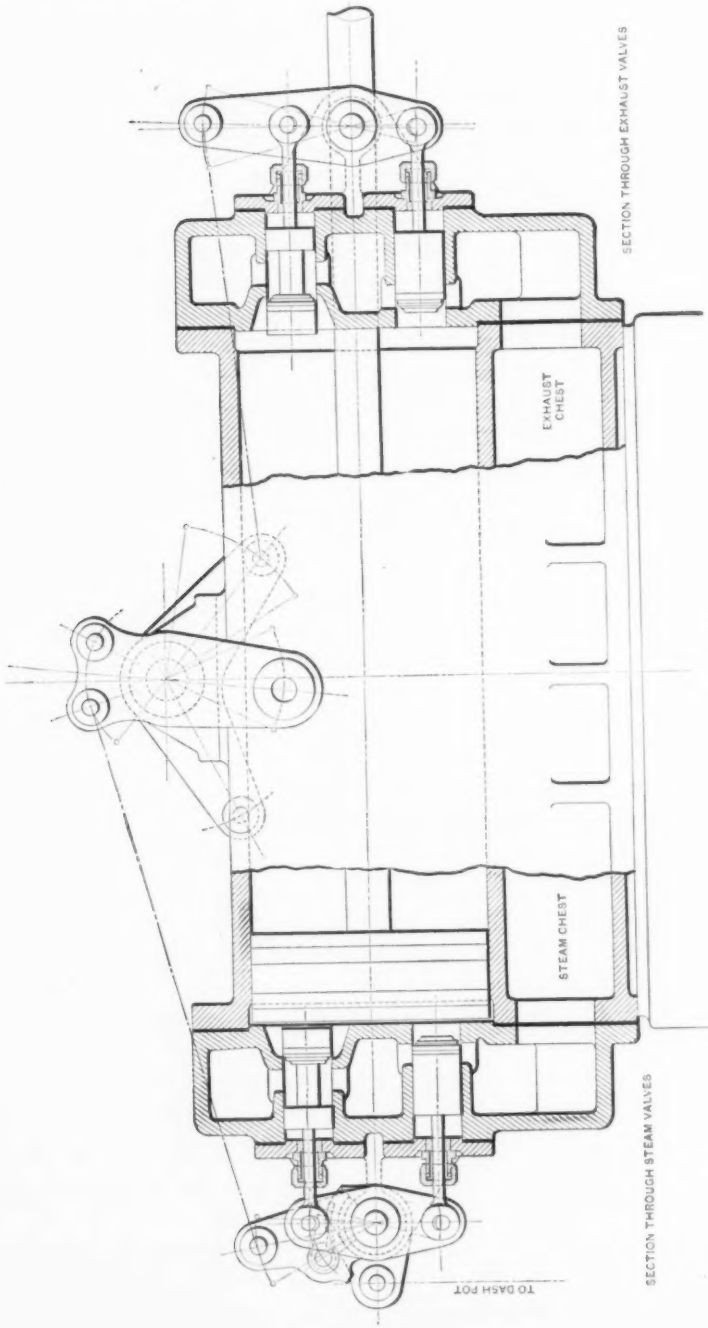

SECTION THROUGH EXHAUST VALVES
EXHAUST CHEST
STEAM CHEST
SECTION THROUGH STEAM VALVES
TO DASH POT
Gordon

FIG. 326

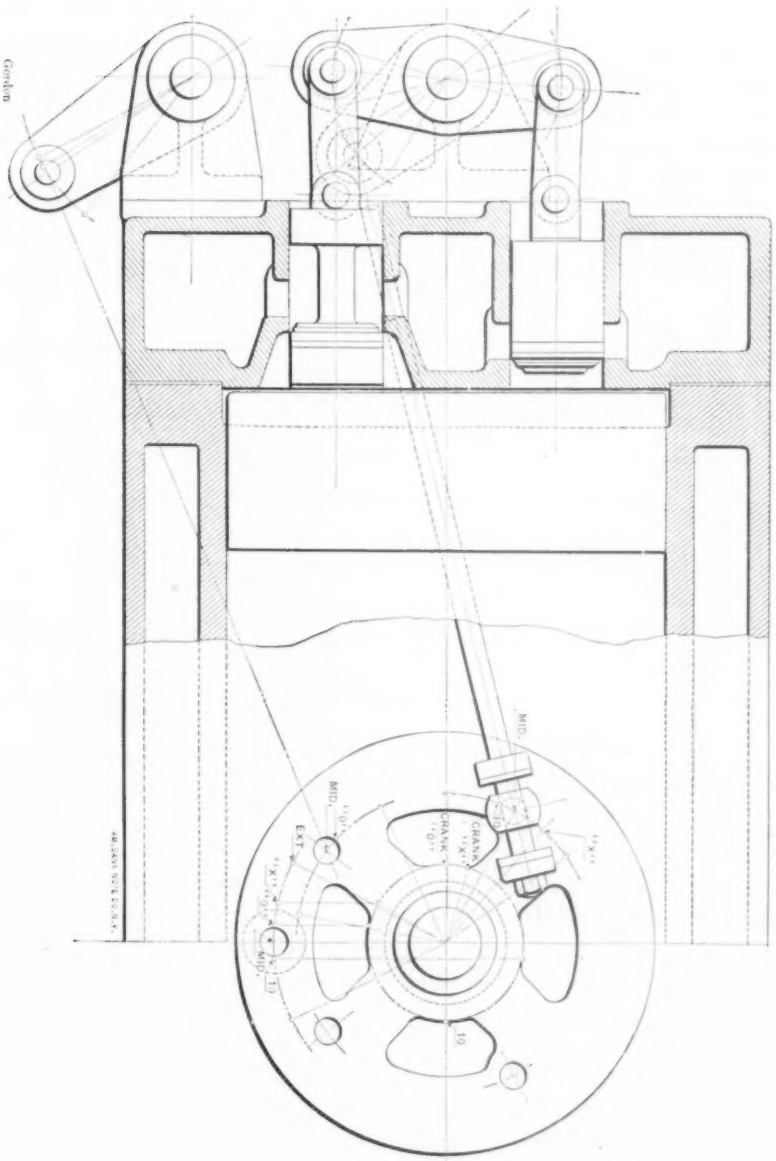


FIG. 327.

diameter by 30-inch stroke. Two of these set vertically are to be driven by gas engines, fed by the waste gases from the blast furnace. To effect economy with gas engines high rotative speed is important, and, as these engines were coupled direct to the gas-engine shaft, they have to run 160 revolutions per minute ; hence the enormous size of the valves, to be made more than ample for 800 feet piston speed per minute. Each of these valves is 11 inches in diameter, two for inlet and two for outlet. Two separate eccentrics operate these valves through the intervention of rock arms and wrist plates. The diagram upon

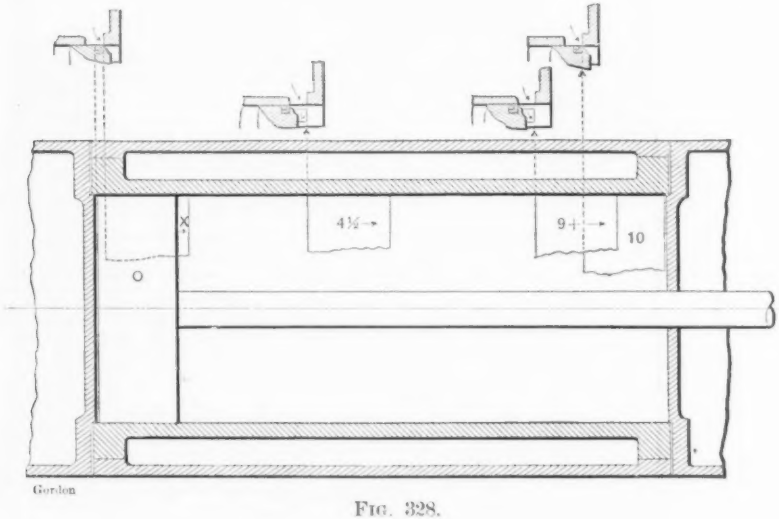
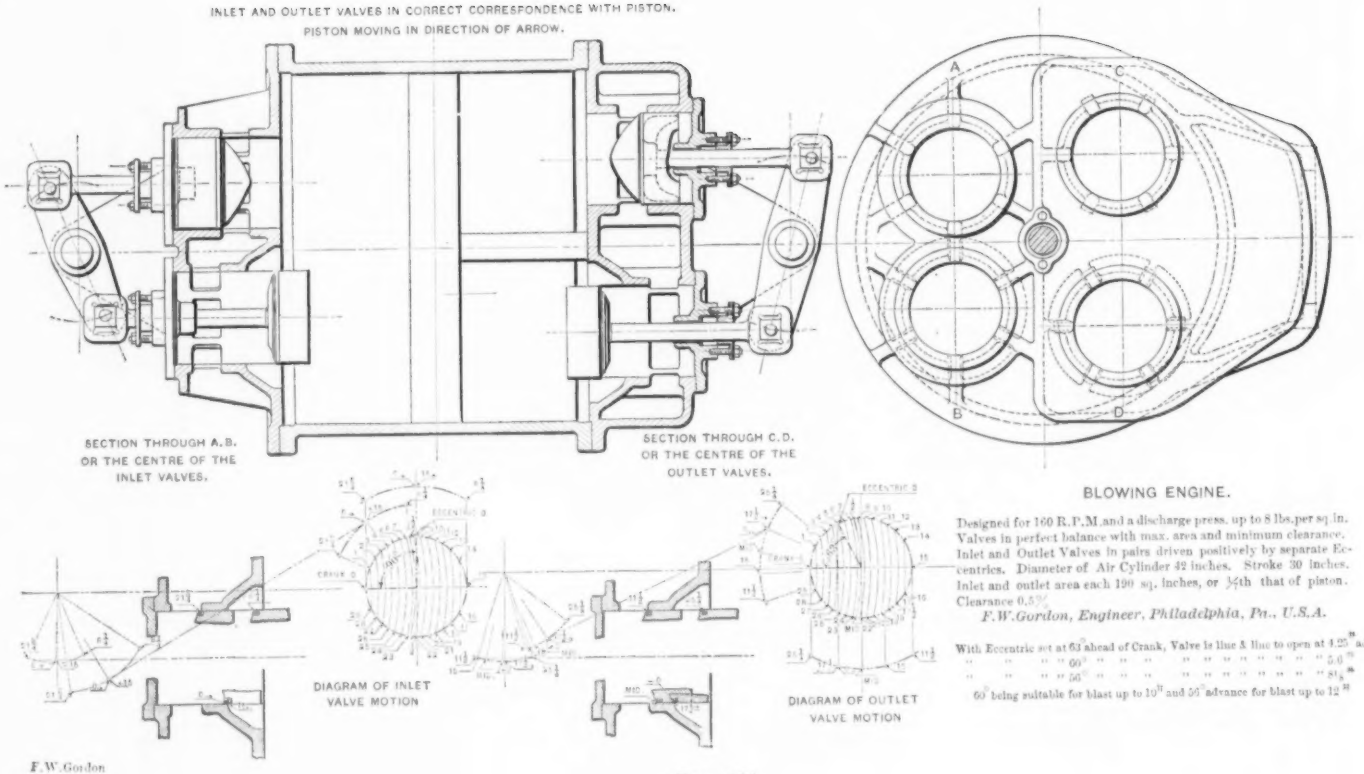


FIG. 328.

the drawing shows the position of each eccentric for taking air from the atmosphere and discharging it at 8 pounds pressure. Each eccentric being positively connected to the valves which it is to operate, leaves the matter of the extreme velocity of 160 revolutions per minute merely a question of large wearing surface and good workmanship. It is, however, apparent that the discharge in this instance is fixed. When the pressure within the cylinder reaches 6 pounds above atmospheric pressure, then the discharge valves are line and line to open. Referring to the action of the automatic discharge valve, it will be seen that this early opening is a decided advantage rather than an objection, and that being line and line at 6 pounds pressure, the valves would compress air economically to 10 pounds ; in fact more economi-



BLOWING ENGINE.

Designed for 160 R.P.M. and a discharge press. up to 8 lbs. per sq. in. Valves in perfect balance with max. area and minimum clearance. Inlet and Outlet Valves in pairs driven positively by separate Eccentrics. Diameter of Air Cylinder 42 inches. Stroke 30 inches. Inlet and outlet area each 190 sq. inches, or $\frac{1}{15}$th that of piston. Clearance 0.5%.

F.W.Gordon, Engineer, Philadelphia, Pa., U.S.A.

With Eccentric set at 65° ahead of Crank, Valve is line 5 line to open at 4.25" adiabatic comp.
" " " " 60° " " " " " " " " " " " 5.0" " "
" " " " 55° " " " " " " " " " " " 5.5" " "
60° being suitable for blast up to 10" and 55° advance for blast up to 12"

FIG. 329.

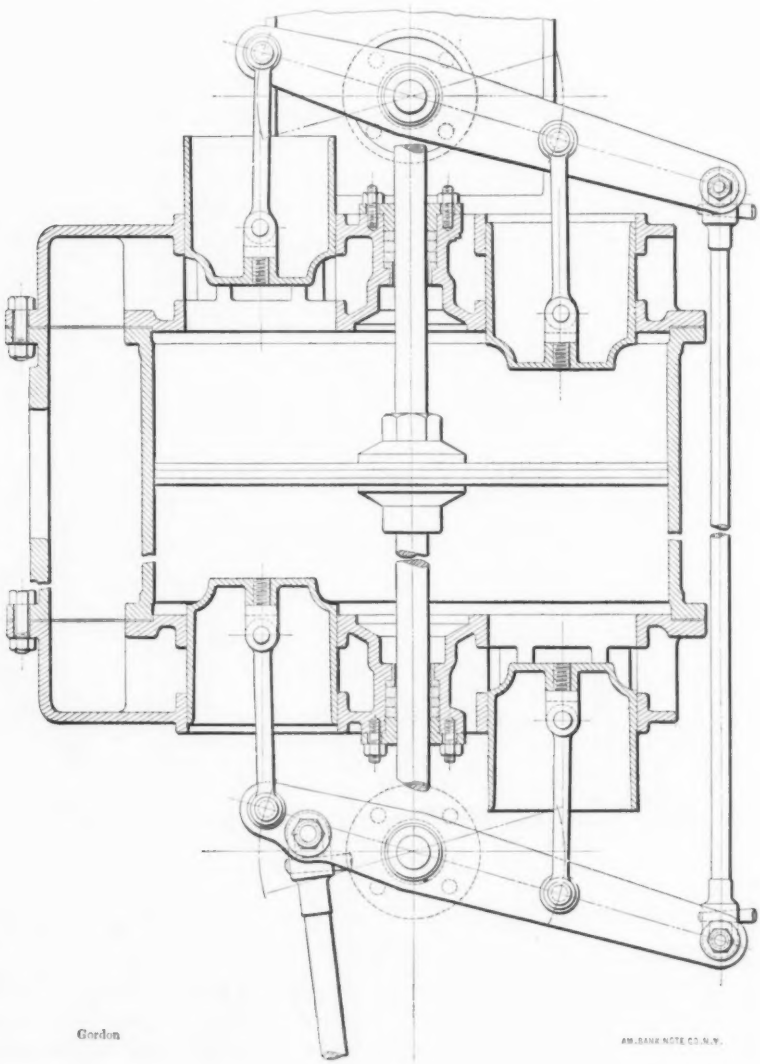


FIG. 330.

cally than if the valves were set to be at line and line when the air reached 10 pounds, or the discharge pressure intended.

Figs. 330, 332, and 333 show the same valves employed for vertical piston blowers, so named as they are expected to supplant the rotary blowers used up to, say, pressures of 4 pounds per square inch. They are simple piston blowers with but two

valves in each head, one of these valves being used for the inlet and the other for the discharge, and all four valves driven by a single eccentric. The valves are always in balance as to pressure and weight of parts, so that the speed at which this machine can be operated simply depends upon the construction of it and the area of the valves. In this instance each valve is made 12 inches in diameter for a piston 42 inches in diameter, giving an area of $\frac{1}{12}$, and they are intended for a piston speed of 600 feet

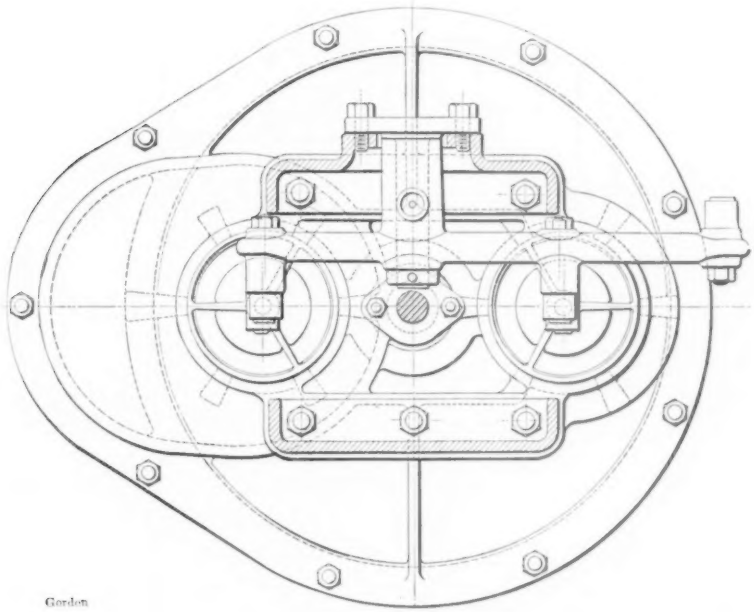


FIG. 331.

per minute. The valves may or may not be metallic packed, depending upon the pressure for which they are to be used. These engines may be set in series, so as to perfectly balance each other as to the weight of parts, and be driven at very high rotative speed. Those shown are intended for 200 to 250 revolutions per minute. They can be depended upon for tightness as completely as the best steam engines. The rotary blower is particularly faulty in this. It is difficult to measure the waste by leakage in rotary blowers; the best effort the writer has been able to make was in calculating the quantity of air by their displacement used to the consumption of a pound of coke in

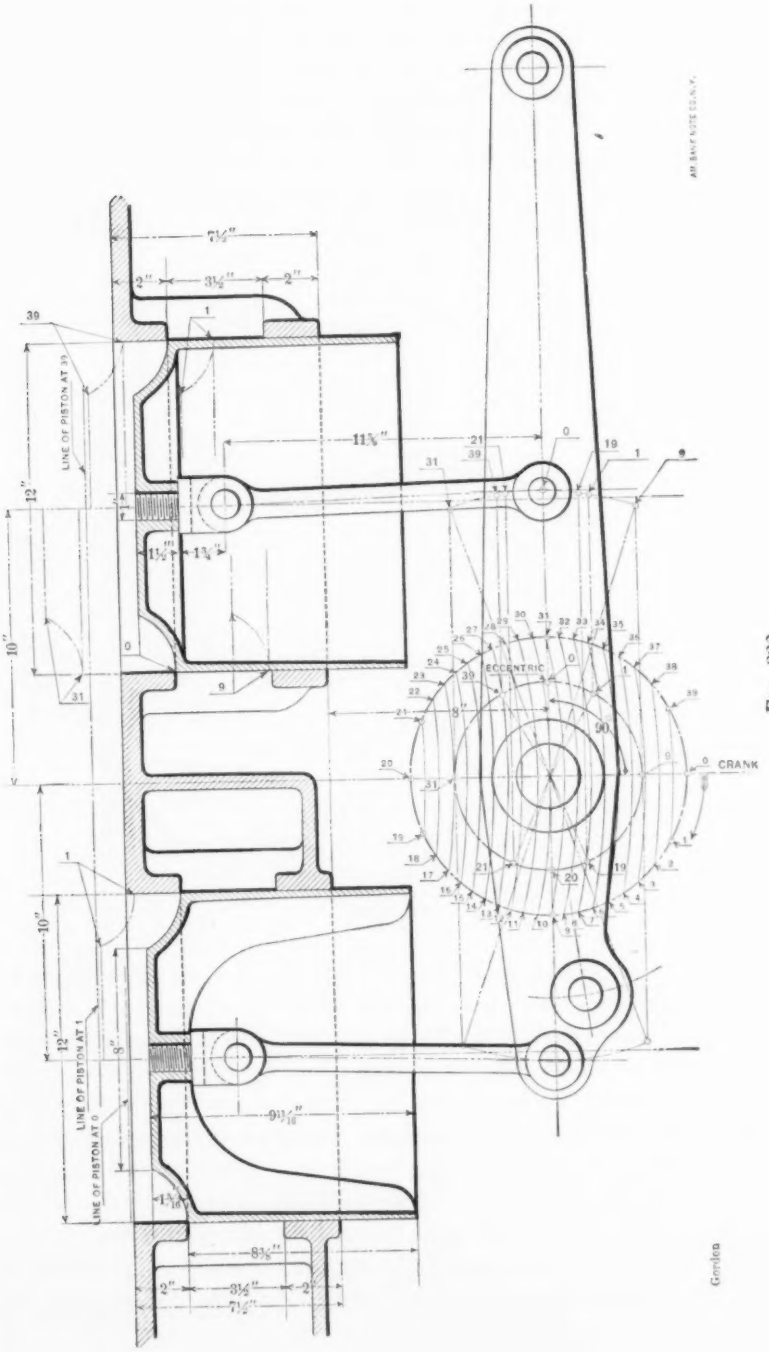

7½"
2"
3½"
2"
39
1
LINE OF PISTON AT 39
11⅞"
21
39
0
19
1
31
12"
1½"
1⅜"
10"
0
31
9
ECCENTRIC
8"
90
CRANK
20
31
19
18
17
21
20
19
10"
12"
8"
LINE OF PISTON AT 1
LINE OF PISTON AT 0
9 11/16"
1 5/16"
8 5/16"
2"
3½"
2"
7½"

FIG. 332.

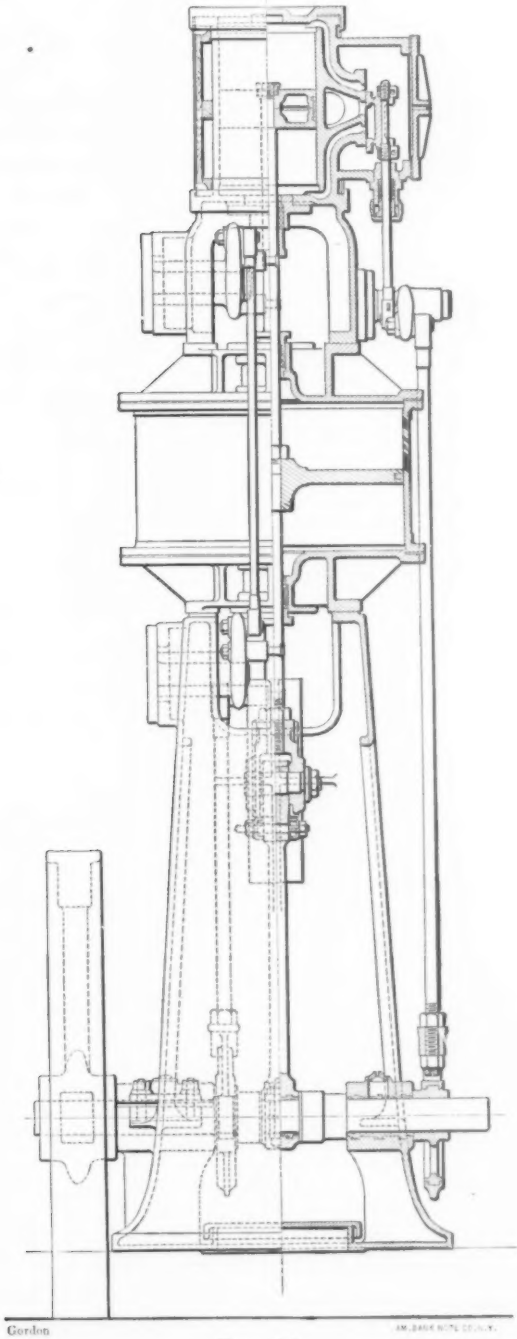


FIG. 333.

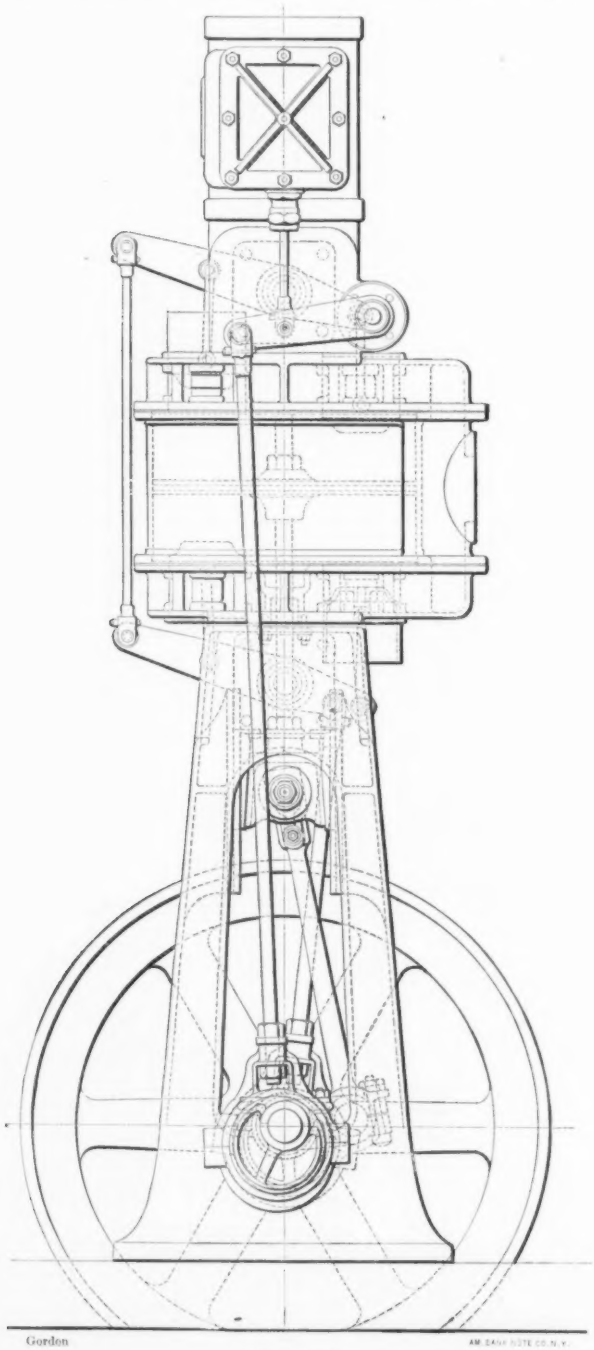


FIG. 334.

cupola practice, where he found as high as 112 cubic feet required by them, whereas in blast-furnace practice, with modern blowing engines, 56 cubic feet is found ample. Equating the two operations on a liberal basis shows that this particular blower was delivering but 60 per cent. of its volume, and this at a pressure of only 12 to 13 ounces. The necessity for tight blowers, where a pressure of 3 to 4 pounds is used, as in silver smelting and pneumatic transmission, is apparent. The design submitted is especially intended for this class of work.

Fig. 336 is a delivery valve for an air compressor, closed by the action of the eccentric and opened automatically.

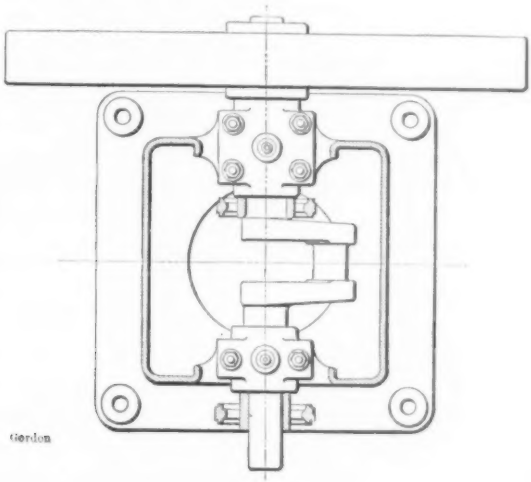


FIG. 335.

The photograph is of an air compressor with inlet valves to drawing No. 1, and an outlet valve to this drawing, No. 9. This valve is closed to line and line by the arm shown, operated from the wrist plate shown on drawing No. 3. It is a piston valve, packed with metallic packing. The valve proper (*A*) is bored to fit stationary piston *B*, *B* being a part of the casting forming the cover of the valve chamber. In the face of the valve *A* is a ball poppet valve *C* opening to the main cylinder. The stem *D* screwed into the face of the valve *A* carries the guide *E*. A short, stiff spring is provided, against which this guide presses immediately before the valve closes to line and line, and when this spring is fully closed it exerts a pressure of about

two-thirds of that which the air would upon the annular space of the valve surrounding the cylinder *G*. On starting without pressure in the discharge chamber, the wrist-plate rod when near toggle easily overcomes this spring. In a few strokes the pressure in the discharge chamber reaches the working pressure,

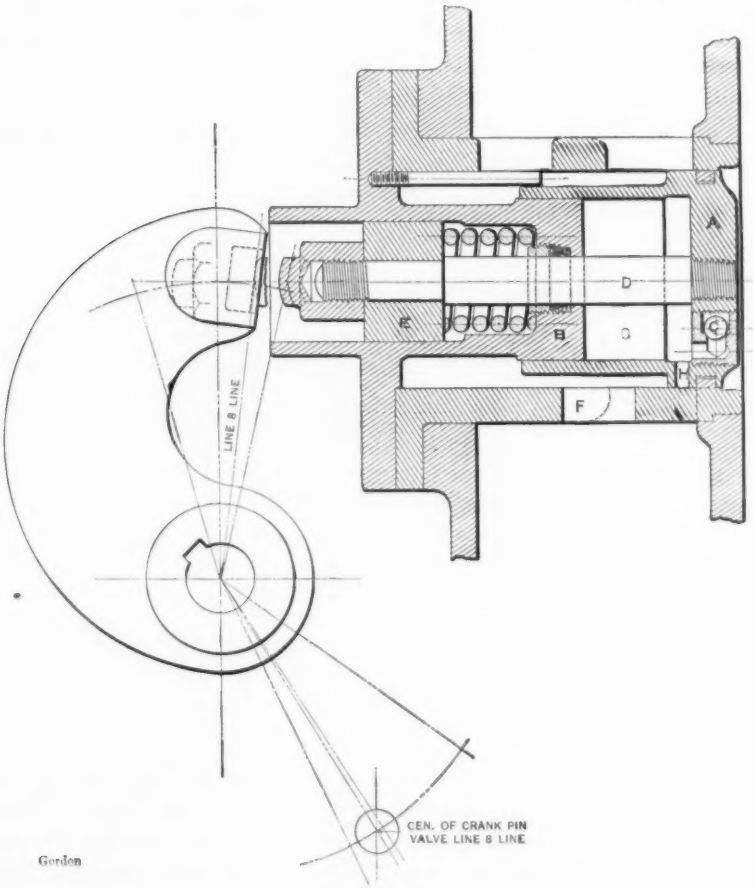


FIG. 336.

when the eccentric or eccentric gear has no resistance to overcome, except the mechanical friction of the valve. The small hole *H* ($\frac{1}{4}$ inch for a 4-inch valve) plays upon the rib and opens to slot *F*. Operation: The valve closes to line and line upon the piston reaching the dead point at the valve end of the cylinder. As the piston recedes, the small volume of com-

pressed air in the clearance is quickly expanded, and the valve flies to full lap, compressing the spring. At the same moment the valve C opens and the pressure in the valve cylinder G is reduced to that of the suction side of the piston. H being closed and ball valve seated, while the returning piston compresses the fluid in the cylinder, the cylinder G still remains at suction pressure, so that the valve as to moving right or left will be in equilibrium (neglecting the spring) when (area $G \times$ suction pressure) + ($A - G \times$ discharge pressure) shall equal the total pressure on A, and the valve will open automatically when this latter pressure exceeds the sum of the two former pressures sufficient to overcome its friction.

Card V-1 is taken from cylinder G with atmospheric suction and 41 pounds reservoir pressure, the indicator drum being driven from the engine's crosshead when the compressor was tandem to the steam engine. Card A is the accompanying indication from the main cylinder of the compressor, both taken with equal weight of springs, and simultaneously, 85 revolutions per minute. The pressure at x, Card V-1, when the valve commences to open, is 16 pounds gauge, whereas, area $A - G =$ 3.28 square inches, area $G = 9.28$ square inches, area $A = 12.56$ $\frac{3.28 \times 55.7 + 9.28 \times 16}{12.56} = 26.4$ absolute. This point is marked o on the Card A, the opening of the valve commencing at x, and it is full open at y. On Card A there is marked the corresponding positions of o, x, and y. It will be seen that there is no indication of the valve having opened at x. Indeed, no air entered the cylinder from the discharge chamber until the valve was wide open at y, else the compression line w would have been increased. The increase in this line takes place after the valve is fully open and before the pressure in the cylinder has reached the pressure of the discharge chamber to the extent seen upon Card A by the increase directly opposite z, an insignificant increased resistance to the piston's movement. This can only be realized if the time occupied during the passage of the piston from x to y is appreciated: Distance on card .26 of an inch; total length of card 4 inches; x to y $\frac{1}{15.4}$ of card, or stroke of piston $\frac{30''}{15.4}$, say 2 inches. The mean travel of the piston is 425 feet, speed between x and y 667 feet, per min-

ute; time required from x to y $\frac{1}{60}$ of a second. The air had scarce time to start to flow in before the pressure within the piston had reached the pressure at y, when the valve was wide open. An ordinary indicator was used in taking Card V-1.

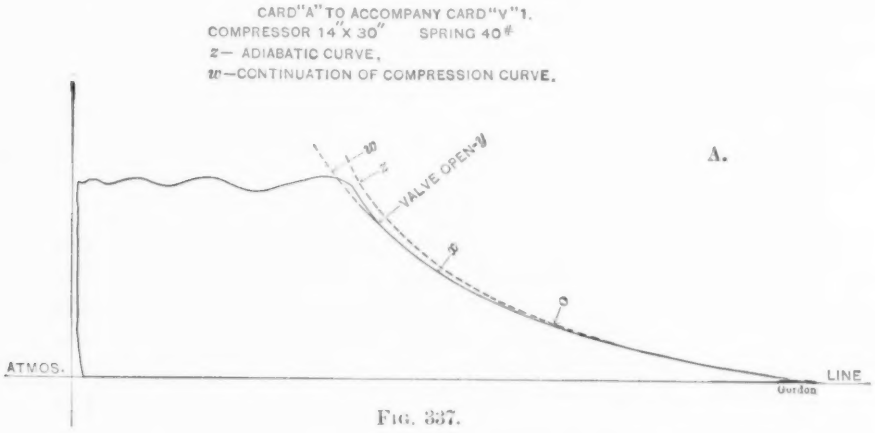


FIG. 337.

Pipe connections thereto were made as short and small as thought advisable, yet owing to the small volume of the cylinder G the clearance added was proportionally large, somewhat changing the pressures as shown on Card V-1 from those which they would ordinarily be with no clearance. As the

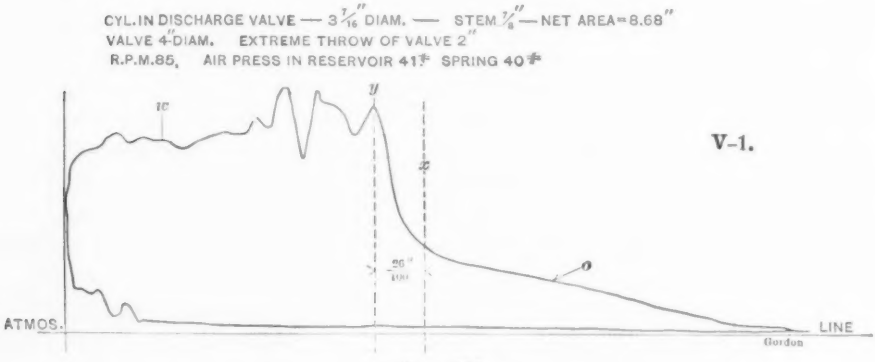


FIG. 338.

valve uncovers its lap, the pressure in G gradually increased—see line o. This movement is due to the strength of the spring, aided by the accumulating pressure in the main cylinder; losing the aid of the spring (which at this point is extended) the rap-

idly accumulated pressure causes the valve to fly open as at x to y. Between these two points, or rather shortly after passing x, the hole H runs over slot in bridge, forming a communication between G and discharge chamber, but this hole is too small to

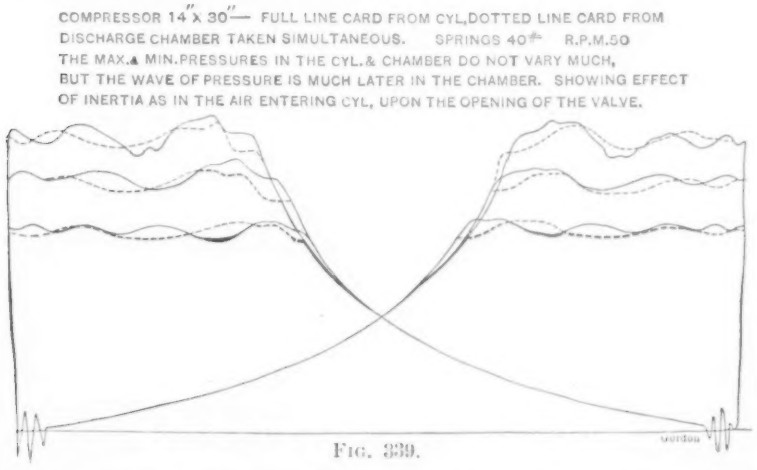


FIG. 339.

permit a free escape, and hence a good cushion is provided, while the valve has no tendency to rebound. The air pressure at w in G in the discharge chamber and in cylinder being alike, the eccentric driven arm, closing the valve line and line, has no force to exert except in overcoming friction. This dead lap on

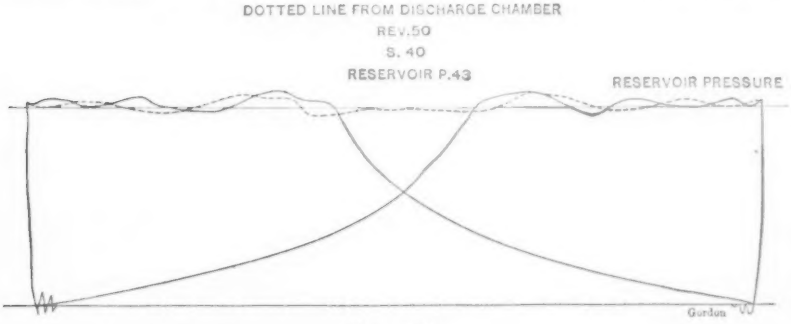


FIG. 340.

a 4-inch piston valve is $\frac{1}{2}$ inch; full movement of valve, 2 inches; $1\frac{1}{2}$ inches of opening. There are three ribs, each $\frac{7}{8}$ inch wide, thus providing 14.91 square inches for the 12.56 actual area of valve. The excess of area is to cover the lack of efficiency of area of the three circumferential ports to the one central

4-inch valve cylinder and to provide for an easy change in the direction of the currents.

The added clearance to the main cylinder by the valve cylinder G is 0.15 per cent. From examination of all the cards taken

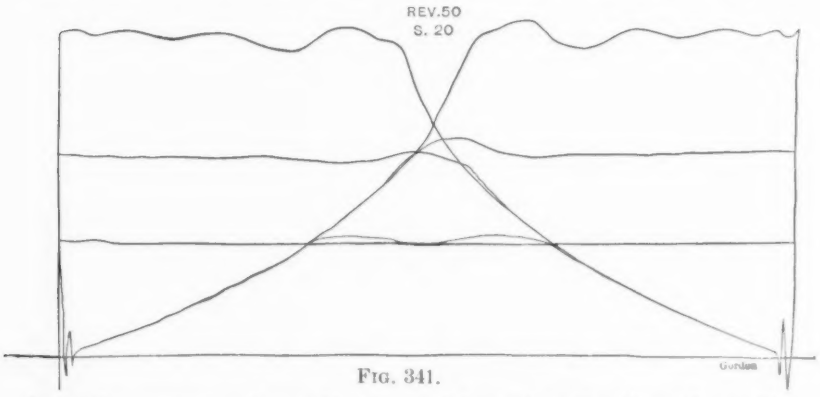


FIG. 341.

from the compression cylinder, and knowing the time these discharge valves open, it is apparent this $3\frac{7}{16}$ valve cylinder G is too large for the highest efficiency—a 3-inch or $2\frac{5}{8}$-inch would be better; but it was sought to open the valve too soon to demon-

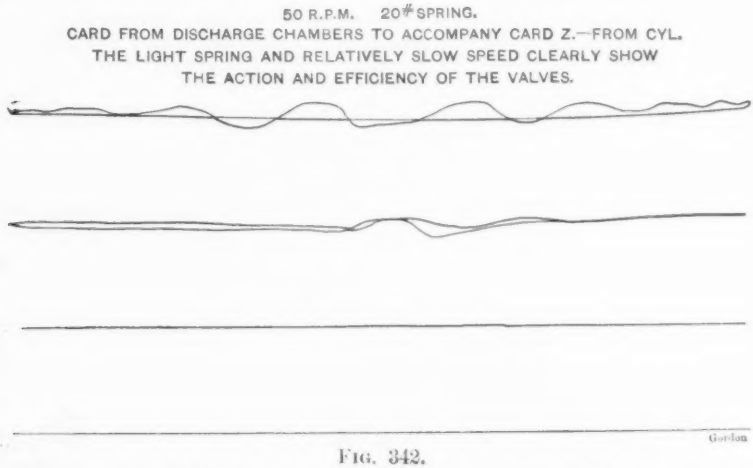


FIG. 342.

strate the feasibility of the system; it is easy to return to a late opening valve.* We are gratified to be able to state that at 100 revolutions per minute (the highest speed this machine was

* See Appendix.

operated) the entire system was practically noiseless—one could simply hear something going on. An automatic poppet valve was tested. This valve was provided with a 3-inch cylinder

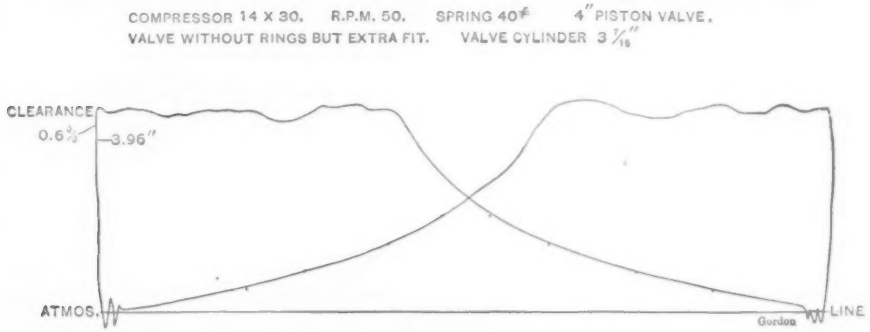


FIG. 343.

similar to *G*. The valve face presented to the increasing pressure in the compression cylinder was reduced by the bevelled surface of the poppet seat, and better result obtained as to time of opening. The cards are nearly as good as the positively

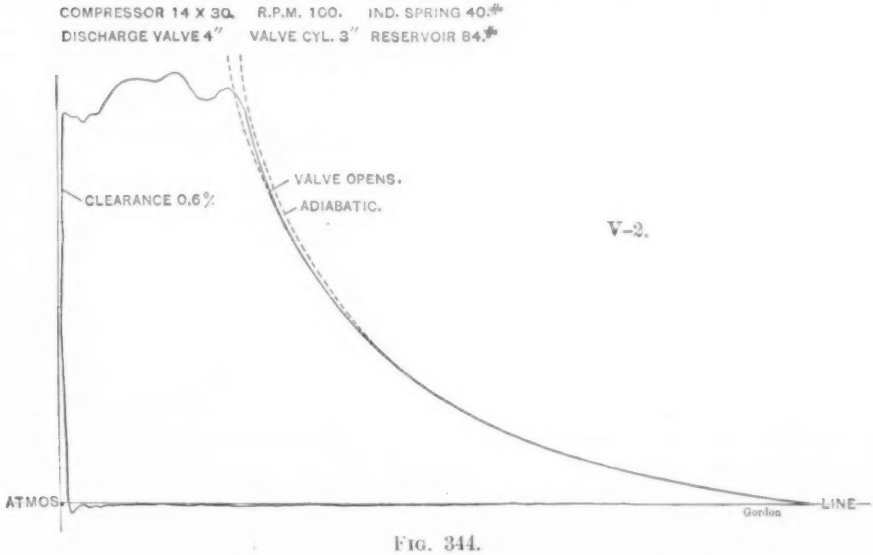


FIG. 344.

closed valve. Springs used to seat ordinary poppet valves are a source of resistance to the outgoing air. The strength of the spring used on this automatic valve was more than neutralized by the introduction of the 3-inch cylinder, and the removal of the

pressure therein, causing the valve to open freely, and checking the fluttering common to such valves.

Card V-2 was taken with these automatic valves, the positive inlet valves before referred to being used in all cases.

FIG. 345.

Inlet Valves.

The cards exhibited show excellent induction line, the atmospheric and suction line being practically coincident. The 14-inch by 30-inch compressor has two 3-inch valves in each head, the actual area being 13 square inches; ratio to cylinder area, 1 to 11—a free inlet under ordinary conditions of eccentric or wrist-plate action, but not such as to give the cards exhibited.

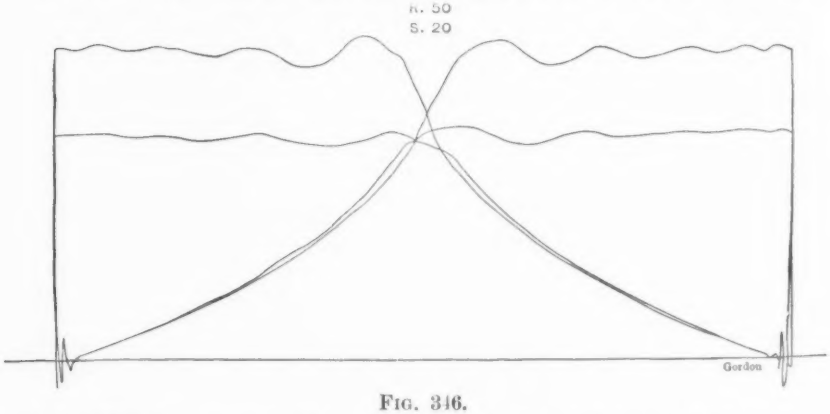


FIG. 346.

Fig. 328 illustrates the action of these valves. They commence to open when the piston has travelled ½ inch, are wide open when it has travelled 13 inches, and remain wide open till the piston has travelled 27 inches of its 30-inch stroke and close at dead point. This compressor is set tandem to a 12-inch by 30-inch Corliss engine. Only one eccentric is used, set at 102½

degrees ahead of the crank, as compressor engines require but a minimum of steam or exhaust lead and but little compression. The compressor's wrist plate is driven by a rod from the steam-cylinder wrist plate (see Fig. 327), but in each of the rods to

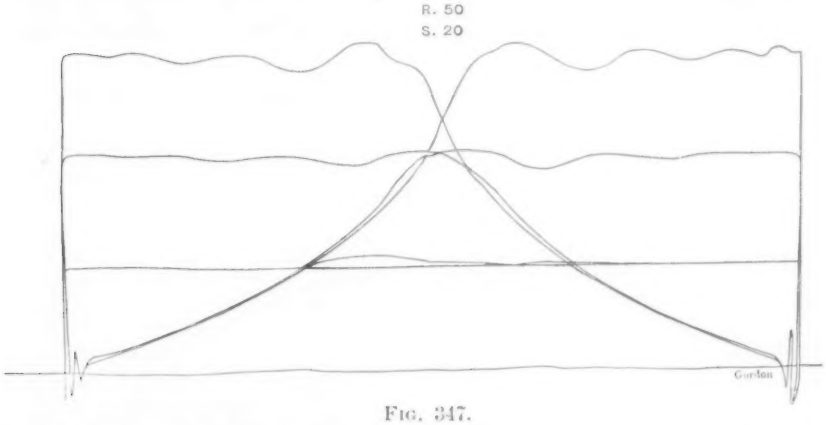


FIG. 347.

the inlet rock shaft there is 1¼ inch lost motion, leather-faced tappets moving the free-moving balance valve—a scheme at once economical as to cost and parts, accomplishing the desired result.

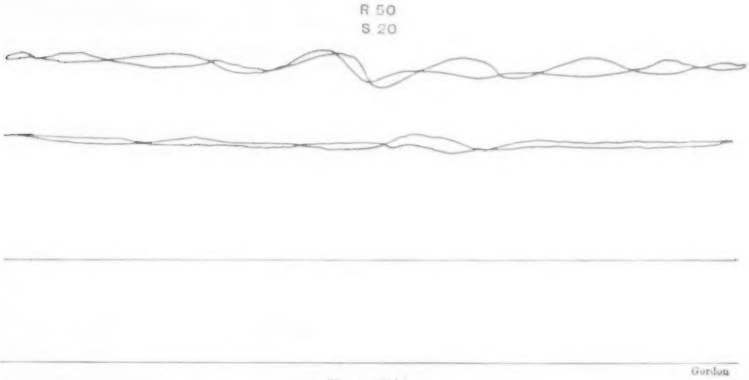


FIG. 348.

Efficiency.

A compressor can only press forward the weight of air received, and among the most desirable qualities is good induction, which, coupled with small clearance, obtains high volumetric efficiency. The total clearance, including that of the outlet valves referred to, is 34 cubic inches on each end (the piston-

head clearance being $\frac{1}{16}$ inch $-$) $= 0.75$ per cent. In compressing to 50 pounds gauge, this volume of air would expand adiabatically to 98 cubic inches in dropping to atmospheric pressure. $\frac{98 - 34}{154} = 0.42$ inch of stroke required, leaving $\frac{29.58}{30} = 98.6$ per cent. as the effective portion of the stroke. If atmospheric

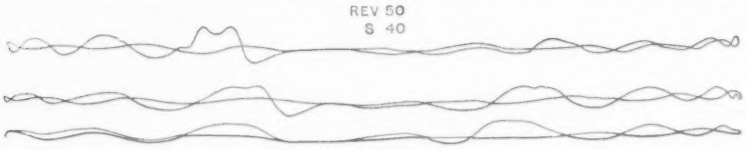


FIG. 349.

pressure and temperature exist in the cylinder at the head of the suction stroke, this will be the volumetric efficiency of the compressor. Up to 60 revolutions per minute the atmospheric line and suction lines on the cards, at the end of the suction stroke, are so nearly coincident that they cannot be seen to differ, while at 100 revolutions per minute the closest scrutiny

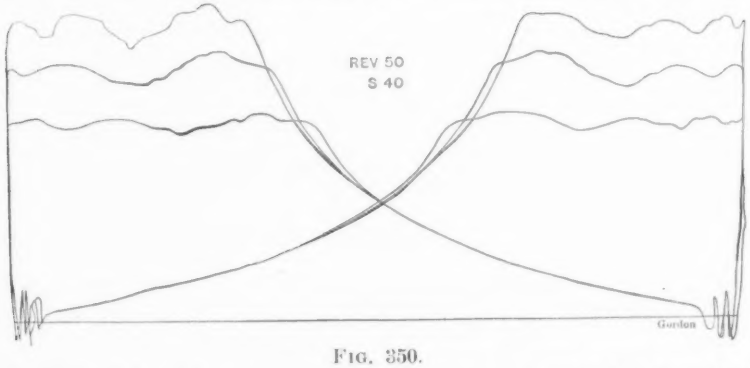


FIG. 350.

is required to note a perceptible drop. The cylinder barrel is well water-cooled, and, with water below the air temperature, should tend to cool rather than heat the air entering the cylinder, leaving the piston head and cylinder head to heat it somewhat. Dry air, however, is only heated by convection—

too slow a process to have much effect on air while it is filling a cylinder; and if the air enters by short and direct unheated passages, it may be assumed that compression will commence upon air at atmospheric temperature. At least this compressor may be safely estimated to deliver 97 per cent. of the weight of

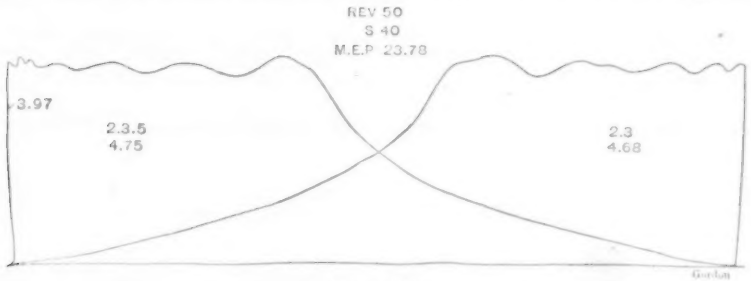


FIG. 351.

its piston displacement of the supplied atmosphere, working up to 50 pounds gauge.

Power Required.

From a number of cards (a few of which are submitted) taken simultaneously from the steam and air cylinders, with springs of same strength, and then changing the indicators and taking an equal number of cards, averaging the whole, the resistance of the air was found to be 87.91 per cent. of the force expended

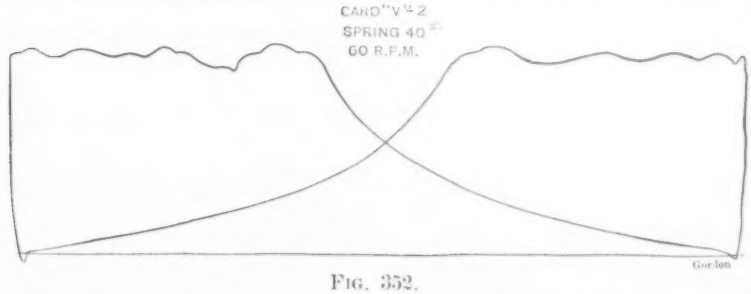


FIG. 352.

by the steam—12.09 per cent. is chargeable to friction; possibly 10 per cent. for large compressors would be an ample allowance. If the connections to the reservoir are large, short, and well formed, the saving in cooling during compression will complete a compression on the equivalent of adiabatic compression in the cylinder. The cards show from 1 to 2 per cent. more power than this, but the connection to and location of reservoir were not altogether the best.

Light Pressure Blowers (*Figs.* 330, 331 *and* 332).

The term "blowers" has become general for compressors first used to supply air to furnaces, cupolas, etc.—that is, for blast pressures now up to 15 or 20 pounds gauge. A modifica-

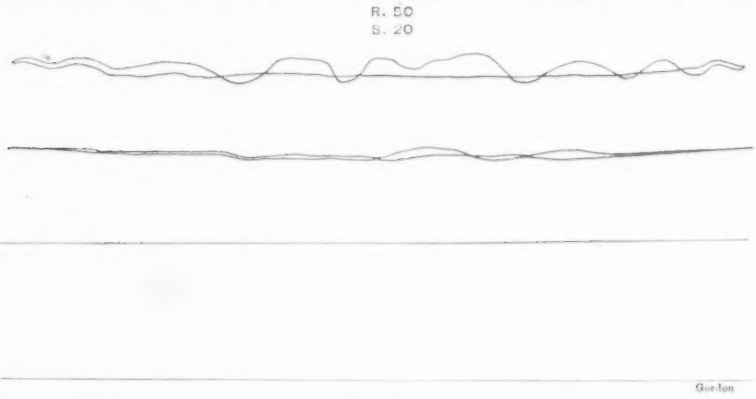


FIG. 353.

tion of the system of our valves is recommended for pressures up to 4 pounds gauge. There are but two valves in each head, both alike, and always balanced, and are positively driven from the same eccentric. One valve in each head serves as the inlet,

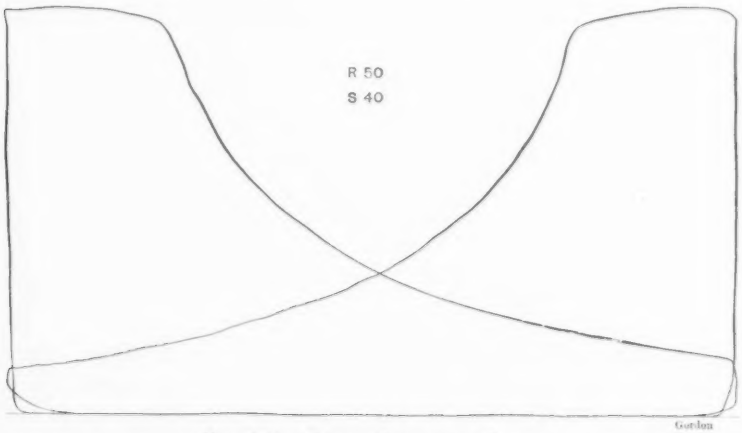


FIG. 354.—STEAM CYLINDER CARD.

the other for the discharge. This is a full-stroke machine, though a slight saving over this may be effected in compressors. The chief merit is the simplicity of the valve action, great portage obtainable, and that it can be driven as fast and main-

tained tight as long as an ordinary steam engine can. While admitting of great portage, the clearance is kept down to less than 1 per cent. for cylinders with a stroke equal to the bore, and a more effective displacement obtained than from ordinary blast-furnace blowing engines. This type of "piston blower" is

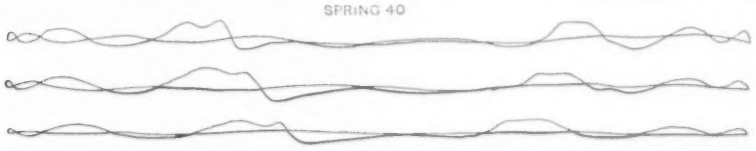


FIG. 355.

adapted to silver smelters, cupolas, pneumatic transmission, etc., etc. A speed of 200 revolutions per minute is well within the limit of this blower.

For higher pressures which are reasonably constant and sel-

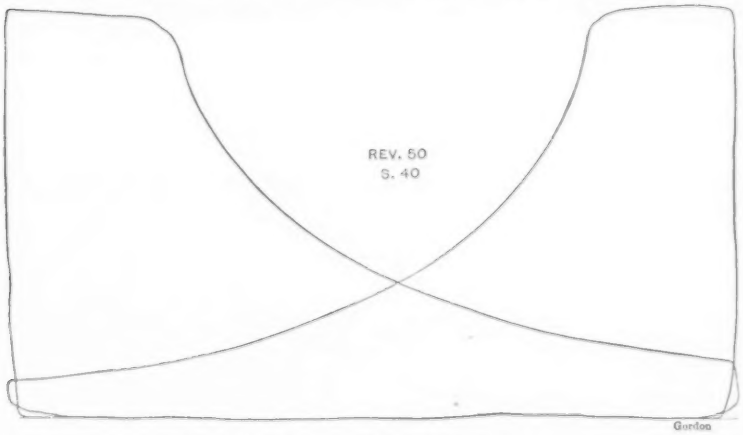


FIG. 356.—STEAM CYLINDER CARD.

dom over 12 pounds per square inch, a blower as shown on Fig. 329 is recommended. There are four valves in each head —*i.e.*, two pairs as in a steam engine, one pair for inlet, the other for discharge. The inlets are driven by an eccentric set about 87 degrees ahead of the crank, the discharge by an eccen-

tric 63 degrees ahead. The latter valves will be line and line to open when the pressure is 6 pounds per square inch in the cylinder; and from what was seen in the action of the automatic discharge valves, if the pressure in discharge chamber were 10

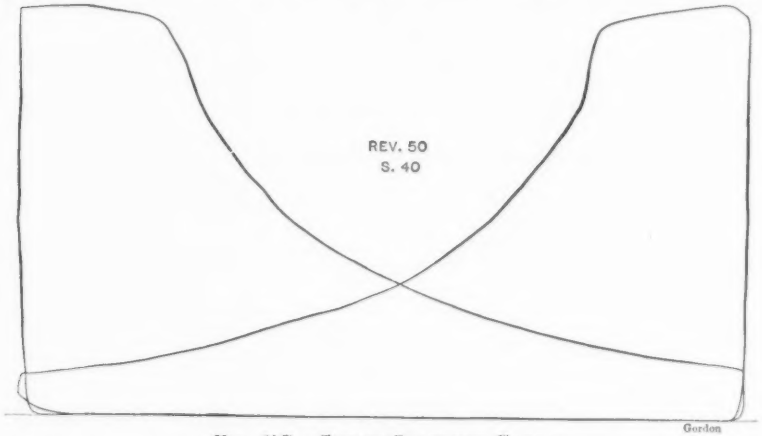


FIG. 357.—STEAM CYLINDER CARD.

pounds an excellent card would result. This drawing was prepared for blast-furnace work, the blowers being driven direct with gas engines at 160 revolutions per minute, and two 42-inch

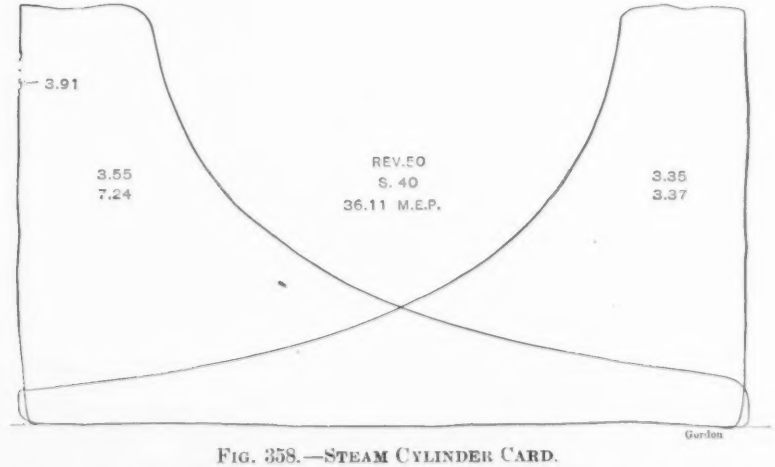


FIG. 358.—STEAM CYLINDER CARD.

by 30-inch cylinders furnishing 15,000 cubic feet of air per minute. The matter is under advisement.

Steam engines driving compressors are subject to double

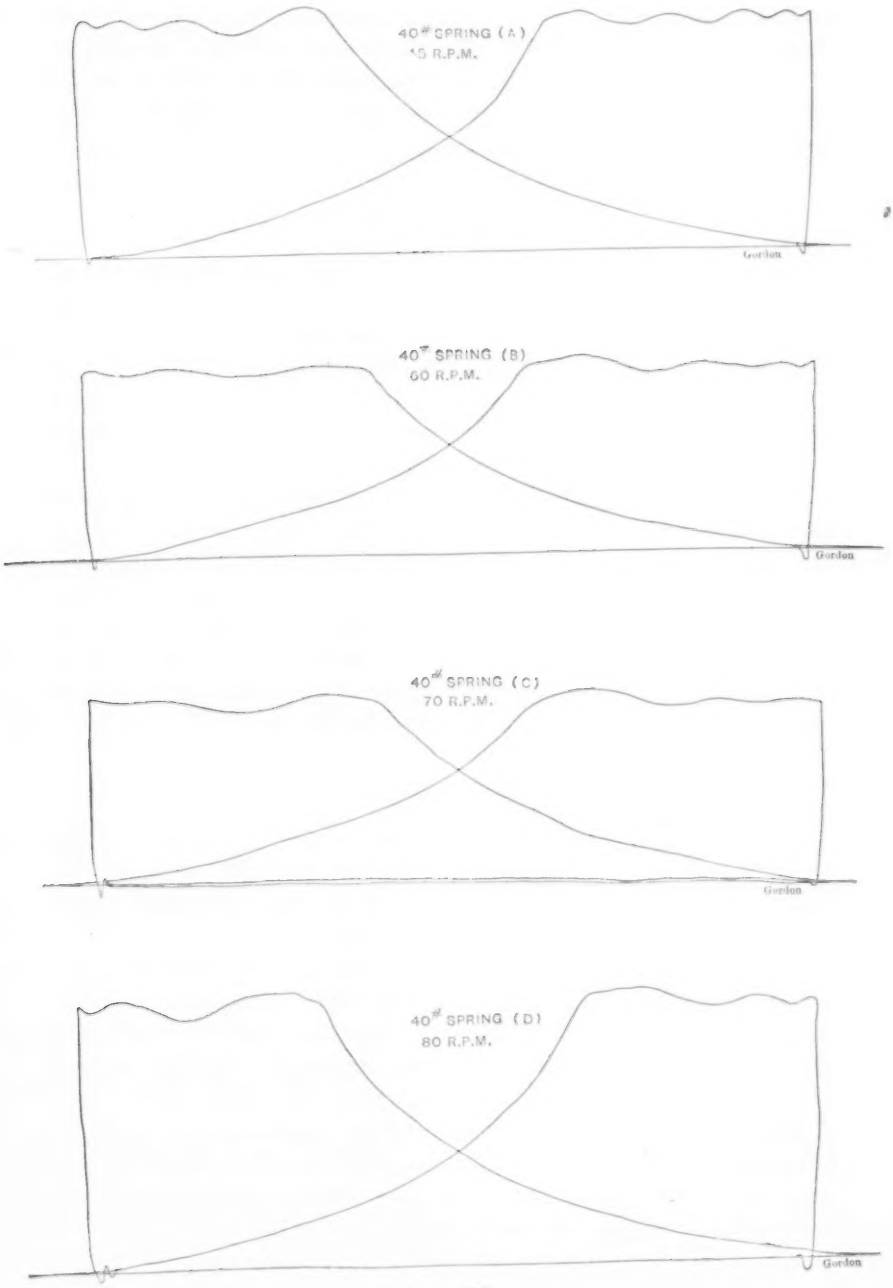

40# SPRING (A)
45 R.P.M.
Gordon
40# SPRING (B)
60 R.P.M.
Gordon
40# SPRING (C)
70 R.P.M.
Gordon
40# SPRING (D)
80 R.P.M.
Gordon

FIG. 359.

strains if they are set with lead of compression. To avoid the loss incident to want of compression clearance should disappear.

With the system of valves herein outlined, less than half of 1 per cent. may be reached, and though no compression is provided, the loss will be trifling.

APPENDIX.

Cards numbered A, B, C, and D (Fig. 359) have been taken from the compressor referred to in the foregoing paper, after the cylinder of the discharge valve had been reduced in diameter from $3\frac{7}{16}$ to $2\frac{3}{4}$ inches. It would appear that the movement of discharge or opening of the valve is much improved, and that throughout a wide range of pressures the opening of the valve regulates itself to the requirements of the situation.

It was a surprise to me that with a very early opening more air did not enter the cylinder, and that a higher average pressure did not prevail in the cylinder by reason of the air entering from the discharge chamber before the cylinder had acquired that pressure. It is proven, both by the action of this valve and by the action in the discharge chamber itself, that inertia is one of the properties of air, however elastic it may seem to be, which pneumatic engineers must keep well before them.

DISCUSSION.

Prof. W. S. Aldrich.—I would like to ask the author about those quick-opening valves, whether he uses the principle of the quick-opening safety valve so as to get the valve open early and wide for the discharge into the receiver?

Mr. Fred. W. Gordon.—I think the action of the valve is made clear by the drawing on page 982, Fig. 336. A $2\frac{3}{4}$-inch cylinder in a 4-inch piston valve deducts from the back or discharge chamber end 6.34 square inches, the discharge pressure acting on only 6.12 square inches, so that when the main piston has compressed the fluid to half that of the discharge the valve is practically in equilibrium, and a small additional force will open it. We all appreciate the importance of early opening of the exhaust valves of steam engines, though at the time of exhaust the pistons are nearly at rest. It would seem to me to be much more important to open the discharge valves of an air

compressor early, as when the discharge is required to begin the piston may be travelling at about its maximum speed, and indeed in a majority of cases will be travelling approximately at that speed. The cards demonstrate the feasibility of this scheme, and show that the pressure at the moment of discharge need not exceed the pressure in the discharge chamber.

DCCCXXV.

MEMORIAL NOTICES OF MEMBERS DECEASED DURING THE YEAR.

HENRI SCHNEIDER.

Mr. Henri Schneider, the son of Eugène Schneider, founder of the works and city of Creusot, was born on December 10, 1840, and passed away May 17, 1898. He became associated with his father in the management of the Creusot Steel Works in 1867, and upon the death of the latter in 1875, although still very young, became the head of this important establishment.

After successful studies at Paris he entered the works under the direction of his father, who desired that he become familiar with all the practical details. Thus he acquired all of his practical knowledge of processes and management.

All his life was a continual activity, in which he found time to think of everything, to associate worldly duties of a very Parisian situation to those of an ironmaster, and to do good judiciously and without counting. From the industrial point of view his work is characterized by a considerable and uninterrupted development of the works of Schneider & Co.

He was the incessant initiator of progress in metallurgy, mechanical construction, artillery, and armor.

In 1876 he perfected processes which made the use of steel possible for armor plates, and in this he remained preëminent. Actively opposed, he put this invention to practical use solely by his perseverance and efforts. After the trials at Spezzia, in which the new steel plates had been shown strikingly superior to those of iron hitherto employed, the Italian navy adopted them on the famous battleships *Duilio* and *Dandolo*.

The Schneider metal was introduced in the French navy in 1881, and its superiority, demonstrated by numerous ballistic tests, caused it to be adopted in the navies of all countries. He was elected Honorary Member of this Society in the end of 1882.

A man of the capacity for work of Mr. Schneider could not remain indifferent to the interests of his country, and he was therefore called to represent his constituents in the French Chamber of Commerce and to defend their interests. He was successively elected councillor of his county, councillor general, mayor of Creusot, and deputy.

He always worked usefully in the interest of his department, and rendered it the greatest services by his high authority.

By his efforts the city of Creusot grew daily, he supplying a system of drainage which had a capacity of 17,000 cubic metres discharge in 1875; gas illumination, and all other works necessary in municipal service. The last donation, given upon his retirement from the mayoralty, permitted the erection of a city hall commensurate with the dignity of the centre of industry which he had led to such a degree of development.

The complete description of the philanthropic work of Mr. Schneider would carry us very far. From the moment of his entry into business he began by an act which revealed the superiority of his ideas—by building the church of St. Charles. He never thereafter ceased his benefactions. He built two workingmen's settlements, one in 1865, the other in 1875; he established gratuitous medical and pharmaceutical service in 1872; a new church in 1883; a home for the aged in 1887, and a hospital in 1894.

The home of the sisters and sick nurses was built in 1897. As one of the first among the great industrials he endowed his employees with numerous and profitable institutions, of succor and of savings; particularly to insure pensions to all employees and workmen, he made from his own resources since 1897 annual donations to the national fund for the support of the aged, which now actually amounts to more than $2,200,000 (11,000,000 francs).

He was made Chevalier of the Legion of Honor in 1867, and Officer in 1878.

Mr. Schneider was regent of the Bank of France, administrator of the Orleans Railway, and of several other industrial societies.

Being corresponding member of various foreign societies of civil engineers, he was made honorary member of the Institution of Civil Engineers in 1889.

In the same year the Iron and Steel Institute, of which he

had been a member since 1876, presented him with the "Bessemer Gold Medal."

FRANK F. HEMENWAY.

Mr. Hemenway was born at Hartford, October 29, 1837. He served a regular apprenticeship as machinist at Keene, N. H., and at the breaking out of the war he enlisted and served throughout. After being mustered out he went to work in the shops of the Troy & Boston Railroad, and was there associated closely with George W. Richardson in perfecting his pop safety valve and the Richardson slide valve. At the time he connected himself with the Society in 1880 he was mechanical engineer and foreman for Starbuck Brothers, Troy, N. Y. It was while connected with this concern that he invented the Hemenway automatic cut-off engine, of which several were built.

He became an appreciated contributor to the *American Machinist* just started, and he became associated with it officially in 1881 as associate editor, becoming editor on the death of Mr. Jackson Bailey in 1887. Mr. Hemenway resigned in October, 1894.

His literary work culminated in the production of his book on the steam-engine indicator in 1885.

The last few years of his life were shadowed by misfortune, so that his considerable mechanical abilities ceased to have their former value to him.

He died in Brooklyn, October 14, 1898.

HORACE B. MILLER.

Mr. Miller's childhood days were spent in Philadelphia. He was born in 1840, and after leaving school in 1854 he began the struggle of life in a modest way. In 1861 he enlisted in the navy and served on board the U. S. S. *Adolph Hugel*. He was in action at Forts Jackson and St. Philip during the opening of the Mississippi. Although enlisted as paymaster's mate, he was really discharging the duties of third officer during these engagements and received the personal thanks and commendation of his superiors.

After the close of the war Mr. Miller embarked in business in New Orleans until 1868, when on account of ill health he was compelled to leave the city. In the same year he took charge

of the office of R. G. Dun & Co. at Toledo, and was successively promoted to offices in Milwaukee and Pittsburg, until in 1874 he was attracted by the field of journalism and became associated with Mr. Baggley of the Pittsburg *Telegraph*. It was from Pittsburg that Mr. Miller went to New York, and in connection with Mr. L. B. Moore and Jackson Bailey and their associates established the *American Machinist*.

In 1895, his health becoming impaired, he was compelled to relinquish his relations with his journal and sought a change of climate, going to southern California with the expectation of building himself up. The benefit seemed but transitory and he passed away quietly at his temporary home at Napa, Cal., October 24, 1898.

By reason of his connection with the business end of engineering rather than its technically professional end, Mr. Miller joined the Society at its outset in the Associate grade. It will be remembered that early gatherings for the formation of the Society of Mechanical Engineers were held in Mr. Miller's office, at that time at 96 Fulton Street, New York.

JOSHUA ROSE.

Mr. Joshua Rose was an Englishman by birth. His father had a shop in the old country, where he learned the machinist's trade, but coming to this country he became a journeyman, working in various places, the longest of which was perhaps his service with the Freeland Tool Works of New York. In the early seventies he began to contribute to the public press, on mechanical subjects, particularly upon shop operations and appliances.

He became one of the editors of the Appleton *Encyclopædia of Applied Mechanics* in 1878. His *Modern Machine Shop Practice* was, however, the work upon which his literary reputation will rest. He was preparing a revision of it at the time of his death, November 14, 1898, at the age of sixty-one years. He became a member of the Society among the earliest charter members, but had practically retired from active labor and was living upon the proceeds of his publications. He was really unable to cope with the labor involved in getting out his new edition and the exertion proved too much for a constitution already enfeebled.

WILLIAM LIVINGSTON CHASE.

Mr. Chase was born November 2, 1855, in West Concord, Vt. He began his work as a mechanic when a boy, in the shops of the Chase Turbine Manufacturing Company, of Orange, Mass., and after having advanced to the journeyman grade he left to enter Worcester Institute, from which he graduated in 1877. Returning to the Chase Turbine Company, he devoted himself to drafting and designing wood-working machinery and water-wheels, and was a joint inventor of a shingle machine. He naturally gravitated into the construction and development of wood-pulp machinery and appliances, and was engineer in charge of construction for the company which preceded the United Indurated Fibre Company, making plates and other moulded ware, and was an inventor of several of their special methods.

The credit has been given to Mr. Chase for being the pioneer inventor for forming vessels from wood pulp at one operation and in one piece.

During the latter years of his life, from 1884 till the time of his death, he was connected with the Knowles Loom Works, of Worcester, having planned the arrangement of the buildings, plant, etc., of the company, and afterwards taking charge of its drafting room and the construction of its regular and new work.

Mr. Chase had become interested in the development of shop systems for making records and keeping costs, and had been concerned in a development of the classification system by the decimal method, upon which he presented a most interesting and valuable paper before the Society at its Chicago meeting in 1893. He was for this reason chosen on a committee appointed to consider the general subject of proposing a proper and more universal system of classification of engineering material by libraries and manufacturing offices, the work of which committee has been interrupted and possibly terminated by his death. He was a passenger on the unfortunate steamer *City of Portland*, which was wrecked off Highland Light in the unprecedented gale of November 27, 1898.

Mr. Chase connected himself with the Society at the New York meeting of 1891.

JAMES FRANCIS.

Mr. Francis was born in Lowell, Mass., March 30, 1840. He was educated in the public schools of his native town until twelve years of age, after which time and until he entered the Lowell Machine Shop as apprentice, in his twentieth year, he was under private tutors. In 1860 he entered the machine shop, and until the breaking out of the war he was familiarizing himself with the details of manufacture of various kinds of cotton machinery. At the first call for volunteers he enlisted, April 20, 1861, and served throughout the war in the Second Regiment of Massachusetts Infantry. His service covered the battles at Cedar Mountain, Chancellorsville, Antietam, and Gettysburg, and he was with Sherman on his march to the sea. He was wounded at Antietam. He rose from the second lieutenancy of the early spring of '61 to the rank of major in 1863, and was commissioned lieutenant-colonel in July, '65, just previous to his discharge.

He became one of the engineers in charge of the Hoosac Tunnel work after the war, but resigned in the following year to enter the service of the Locks and Canals Company, on the Merrimac River, rising in 1867 to first assistant engineer, and in January, '85, to agent and engineer. Again in 1893 he was appointed general agent.

It is difficult to separate the work of Mr. Francis from that of his eminent father, Mr. James B. Francis. It is, however, doubtless just to both to say that the work of either could hardly have been done in its present form without the assistance of the other. Mr. James Francis was, in some directions, less original than his father, but he had a most profound respect for methods which had proved successful. He was tasteful, painstaking, and accurate, and devoted himself to his profession and public interests in the town in which he lived. It has been said that the very loyalty with which he held to what he believed to be his duty during his father's life restrained him from writing for publication on technical matters, and from taking public position.

He connected himself with the Society on the proposition of his warm friend, the late J. C. Hoadley, at the Washington meeting in 1887. His death in Lowell followed a period of failing health for some months, and took place December 1, 1898.

SAMUEL McELROY.

Mr. McElroy was one of the veterans of the profession, having been in constant occupation since 1847 in the fields of railroading, water-works, canals, and rivers and harbors. His railroad connections were with the laying out of the Long Island South Side line, the New York, Bay Ridge and Jamaica line, the New York and Sea Beach lines, together with others in and about Brooklyn, Newtown, Albany, and other places. His water-works experience covered those of the cities of Brooklyn, Hartford, Hamilton, Watertown, and others, and his paper before the Society, descriptive of the great pumping engine of Brooklyn, was a leaf from his experience in this department. His canal experience covered work upon the Erie, Champlain, Chimbote, Oneida Lake, and upon the Niagara ship canal, and many others.

It is perhaps from his connection with hydraulic engineering as concerned with river and harbor work that Mr. McElroy was best known. It brought him into connection with work upon the Hudson River improvements, the Kent Avenue Basin, the Bay Ridge docks, and into relation with public works at the Brooklyn Navy Yard, in the Wallabout, and elsewhere. So wide an experience necessarily brought into his office the erection of buildings of some importance, both engine houses and governmental structures. He had also served as a mechanical engineer in many of his engagements, and was at one time assistant engineer, U. S. N., serving upon some of the vessels whose names are now but memories and whose engines have become historic.

Mr. McElroy was also an inventor in the hydraulic field, and his name attaches to an open-throat hydrant nozzle. In the latter years of his life his work has been mainly that of the expert in various railway bridge and hydraulic cases. He died December 10, 1898.

HILLARY MESSIMER.

Mr. Messimer was born at Pottstown, Pa., October 7, 1833. His apprenticeship began in the shops of the Reading Railroad in 1850, where he served five years, and then went out with the expedition to raise the vessels sunk in the harbor of Sebastopol. In 1859 he entered the Engineer Corps of the United States

Navy, and served with distinction through the Civil War and until 1867. He was on the flag-ship *Wabash* during the memorable naval engagement at Port Royal, and was the first man to go ashore after the capitulation of the forts. This experience made him at one time shipmate with Dewey, Sampson, and Watson. After the war and until his resignation he was with the Mediterranean Squadron.

In 1868 he again entered the employ of the Reading Railroad and became master mechanic of the East Pennsylvania Division, and later of their main shops, where he superintended the building of the marine engines for certain steam colliers. He then went to England to carry on experiments in burning anthracite on war vessels, but returned in 1874 and remained with the Pennsylvania and Reading Railroad, under Messrs. Gowen and Wootten, engaged in various service, such as introducing anthracite into South America, and in the construction of railroads in Brazil and Bolivia, and in experimenting upon the use of inferior fuels in France, Italy, and Switzerland.

In 1882 he became mechanical engineer for Coxe Brothers at Drifton, where he had charge of the machine shop, foundry, boiler shop, etc., and charge of the machinery of all the breakers and buildings used for the preparation of coal. In 1887 he resigned to start the Kansas Bolt and Nut Works, but remained only two years, when he accepted the invitation to become superintendent of motive power and machinery for the Calumet and Hecla Mining Company at Calumet, Mich., with whom he remained until his death at Wornersville, December 10, 1898.

He became a member of the Society at its New York meeting in 1883.

J. C. DEBES.

Mr. Debes was born in Saxe-Coburg-Gotha, Prussia, in the year 1835.

After finishing his technical education in German schools he worked as machinist with Platt Brothers & Co., Oldham, England, and in the Douglas Foundry at Dundee in Scotland, coming to America in 1862. During the war, from 1863 to 1865, he was a draftsman in a government inspection office under Messrs. Alban Steimers and W. W. Wood, and worked under Captain Ericsson on designs of the light-draft monitors such as were used during the Civil War. From 1866 to 1868 he was engaged with the George

H. Corliss Company and with George H. Reynolds, building marine engines at Mystic Bridge, Conn. He moved West in 1868, and connected himself with the C. & G. Cooper Company at Mt. Vernon, Ohio, as superintendent and mechanical engineer. Leaving them in 1872, he became superintendent of the Jackson Foundry and Machine Company of Jackson, Mich. He moved from there to Milwaukee, serving with the E. P. Allis Company on outside engineering, and as superintendent and mechanical engineer for the Weisel & Vilter Company until 1882, when he again returned to the C. & G. Cooper Company, with whom he remained until the day of his death from heart failure, December 13, 1898.

Mr. Debes became a member of the Society at the Atlantic City meeting in May, 1885.

MILAN C. BULLOCK.

Mr. Bullock was born in Grandville, Washington County, N. Y., June 26, 1838. He worked on a farm until, at eighteen years of age, he became apprenticed to a miller; but a tendency to asthma compelled him to abandon that trade, and precluded his service in the army of 1861 after he had promptly enlisted on the declaration of war. At the age of twenty-four Mr. Bullock entered himself as machinist apprentice, and was very early made foreman. In 1866 he was attracted to quarrying machinery, and entered the employ of the Steam Stone Cutter Company, of Rutland, Vt., and afterwards the Windsor Armory Company, where he gained the experience in channelling and diamond drill work which subsequently became of value to himself and to the mining industry in general. In 1870, as superintendent of the Pennsylvania Diamond Drill Company, Mr. Bullock put down the first very deep hole by the use of a diamond drill, at Phœnix Park. It was at this time that he made and patented a number of important improvements, and in connection with Mr. Jacob Shelby brought out and patented the "long-hole process" of shaft sinking. One of his earliest achievements was the sinking of the two East Norwegian shafts near Pottsville, Pa., at rates ranging from 80 feet to 130 feet per month, as compared with the speed of 20 feet per month by the older process.

In 1871 Mr. Bullock accepted overtures from a larger diamond drill company, and his first work was the introduction of the

diamond drill into the work at Hell Gate, in New York Harbor, under General Newton. The diamond drill was used in exploring ahead of the work in tunnels, headings, chambers, and the like. It was Mr. Bullock and his company who furnished the plant in 1873 for driving the famous Lima and Oroya railroad tunnel in Peru through the Andes.

The American Diamond Drill Company failed in 1874, and carried Mr. Bullock down with them financially. He was therefore forced to begin over again, and a year later, in July, 1875, Mr. Bullock moved to Chicago, where he entered in business on his own account with a very limited capital. He built up the business of manufacture of diamond drills and other rock-piercing machinery for mining and prospecting, and organized a prosperous business. He was the introducer into America of the Willans high-speed engine.

He connected himself with the Society at the New York meeting in 1882, and although he had been in failing health for over a year his death came unexpectedly, January 12, 1899.

JOHN R. WAGNER.

Mr. Wagner was born in South Bethlehem, Pa., May 12, 1861. He had the history of so many successful men who have borne the burden in their youth of a necessity for self-support, even in childhood. He began first as a farmer's lad, and afterwards in charge of the news-stand of a railroad station, assuming the care of a widowed mother. He first turned his attention to telegraphing, but his lack of common-school education stood in his way, and he became an apprentice with the Shive Steam Engine Governor Works, of Bethlehem, where he served four and one-half years. He then went to Philadelphia to perfect himself in his trade, taking evening school work in addition. At the age of twenty, while acting as foreman of shop, he prepared himself for the entrance examinations to the School of Mechanical Engineering at Lehigh University, and maintained himself by a loan from influential friends, which he returned within a few years after graduation. After completing his work at Lehigh he was offered a superintendency with the Shive Governor Works, but preferred to accept an offer made by Mr. Eckley B. Coxe, to whom he became general assistant and assistant superintendent of the testing department, both of the

Cross Creek Coal Company and of the Coxe Iron Company. It was in this relation that he became specially expert in coal analysis, and with particular reference to economy in use of coal for steam. He was a successful lecturer in this chosen field, and delivered addresses before such organizations as the Scranton Engineers' Club, and in further presentation of it articles appeared in the engineering magazines over his signature.

The death of Mr. Coxe in 1895 closed this ten-year chapter in Mr. Wagner's life. The experiments on coal were stopped, but in the following April he was made superintendent of motive power of the D., S. & S. Railroad, while retaining his duties as superintendent of the testing department. Tempting offers were made to him from the Anaconda Company of Butte, Montana, and from other firms in Pennsylvania, but he preferred to remain with his former benefactors, and received gradually larger responsibilities until he became in charge of the construction department and full superintendent of motive power and machinery. These positions he held at the time of his death, January 21, 1899, at the age of thirty-seven. His death was caused by pneumonia.

Mr. Wagner organized at Freeland, Pa., a school known as the Mining and Mechanical Institute of the Anthracite Coal Regions of Pennsylvania. This was started for the purpose of providing an education for boys of small means and few advantages. Evening classes are held, and a reading room and reference library are open to ambitious boys. It is interesting to see in this undertaking of Mr. Wagner's the effect of the inspiration of the great Eckley B. Coxe.

GEORGE A. BARNARD.

Mr. Barnard was also one of the early members of the Society, becoming a member in June, 1880. He was born in Albany, N. Y. Previous to 1870 he had had six years' experience in the shop and in marine engineering, and served from 1870 to 1874 with the J. C. Hoadley Company, of Lawrence. In 1874 he became identified with the Buckeye Engine Company, and was with them at the time he connected himself with the Society, in 1880, as manager of their Eastern agency, in connection with Mr. William Lee Church. In his later years he had turned his attention to other problems in power-house practice and in the

lines of steam heating, and had devised and arranged the form of cooling tower which is known by his name. He passed away February 8, 1899, in the fifty-eighth year of his age.

JOHN KRUESI.

Mr. Kruesi was born May 15, 1843 at Speicher, and educated at St. Gall in Switzerland. He entered the mechanical trade at an early age and received a very thorough groundwork in the same in his native country. After working in Paris, Hamburg, Copenhagen, and London he decided to come to America, where he arrived in 1870. Here he married and became an American citizen.

About 1872 he went to work for Thomas A. Edison, who was then engaged, apart from his inventive pursuits, in the manufacture of electric instruments in Newark, N. J. He there showed remarkable ability in devising labor-saving tools and other methods of cheapening manufacture.

In 1876, when Mr. Edison built his laboratory at Menlo Park, N. J., and moved there, Mr. Kruesi went with him as foreman of the mechanical department, and in the next few years won the respect and friendship of his chief by his ingenuity, his ability to grasp the great variety of experiments that he was called on to make, and his untiring energy as head of a shop that was open twenty-four hours a day for pretty near the whole year round.

When about to place the incandescent light in the hands of the public Mr. Edison selected him as a partner in and general manager of a company formed to manufacture underground electric tubing about 1881, and the great perfection of the "Edison underground system" as now used is due to the faithful and able manner in which he carried out the ideas given him during the next few years.

About 1886 the Edison Tube Company was absorbed by the Edison Machine Works, a New York manufactory of dynamos and electric devices owned by Thomas A. Edison and Charles Batchelor, and Mr. Kruesi became the assistant general manager of the whole. This company shortly afterwards bought property in Schenectady, N. Y., and established its factory there, this factory being the nucleus of what is now the General Electric Company's shops.

In 1892 Mr. Kruesi became general manager of the General Electric Company's works, and continued so until 1896, when, owing to the enormous increase in the company's business, it was thought better to have a greater division of labor, and he was appointed chief mechanical engineer, which post he occupied at his death.

It is admitted generally that the plant of the General Electric Company at Schenectady, with its tools and buildings, all of the most modern type, stands second to none in the world, and the credit of this is in a great measure due to Mr. Kruesi, under whose charge the facilities necessary for 300 hands in 1886 were increased to those necessary for upwards of 4,000 hands, which the company has employed during the last few years.

Mr. Kruesi became a member of this Society at its Montreal meeting, June, 1894.

FREDERIC COOK.

Frederic Cook was born in London, England, February 6, 1829. He came to this country with his brothers, commencing his professional life by a service with Stillman, Allen & Co., at the Novelty Iron Works, as apprentice. He passed through the finishing shop, pattern shop, and drawing office, taking his first responsible position as head draftsman at the American Machine Works, Springfield, Mass. He was then twenty-one years of age. In the early forties he went to California, around Cape Horn, to receive and sell out cargoes of machinery, and remained there two years. He then became a partner in the firm of Lange, Cook & Co., and later a member of the firm known as the Belleville Iron Works, located at Algiers, La. Out of that shop came a bagasse burner, invented by Mr. Cook, which is still in service.

On the breaking out of the war Mr. Cook went to Canada, and returning in 1863 opened a consulting engineering office and patent agency in Chicago. From this office Mr. Cook made plans of the Pittsburg grain elevator for Edward Burling. During 1865-7 he was consulting engineer and superintendent of the Wyoming Rock Oil Company of New York and Ontario, and was one of the early engineers to make commercial success of the use of petroleum. He was invited by the Government of

Austria to visit that country and demonstrate its use for their navy.

In 1868 he made plans for the machinery of the Havemeyer & Co. sugar house at the foot of North Second Street, Williamsburg, but was lured back to the South in 1869, where he took up the machinery business and sugar planting. About this time he invented the Arrow cotton tie and became the general agent of the American Cotton Tie Company, of Liverpool, which was organized partly under his open slat cotton tie patent, and was licensed under twelve other of his inventions. He remained with this company eight years, until the expiration of the patent period. He was the inventor of some sixty mechanical contrivances, which brought him in a large income. In 1880 he became the agent for the Babcock & Wilcox Company, and was manager of their Southern office, handling a large proportion of the business of that company in the West Indies.

His relaxation was his devotion to painting with the brush. He became a member of the Society at the New York meeting in 1886, and passed away February 26, 1899.

HIRAM KIMBALL.

Mr. Kimball was born at Randolph, Vt., in 1845. After an early common-school course and a partial course in the Law School of Columbia University, he was admitted to the United States bar in 1880. Previous to this time he had been business agent for the Toronto Rolling Mill Company, and had been for a short time associated with the late A. L. Holley during the early period of the development of the American Bessemer process.

His mechanical instincts, however, diverted him into the field of construction, and he identified himself with a form of rail joint, with the construction of the Chapman jack-screw, and with the Butler Draw-Bar Attachment Company. His principal connection, however, from 1883, was with the turnbuckle department of the Cleveland City Forge and Iron Company. He was at first engineer and later manager. In 1893 he acquired large interests in the Cleveland Street Railway development and in the management of other important business interests.

He connected himself with the Society at the Chicago meeting in 1886. After an illness extending over many months, he passed away March 9, 1899.

HAMILTON A. HILL.

Mr. Hill was one of the charter members of the Society, his application being dated May 7, 1880. He was born in Worcester, Mass., January 2, 1832. After graduating from Harvard College with the class of 1853 he entered the law school, and later the office of the Hon. George F. Hoar, and began the practice of law in Boston. By reason of a severe illness he was compelled to abandon this profession, and for the remainder of his active life was engaged in the business end of mechanical engineering. His specialty was the agency for power plants and manufacturing plants, to which he brought an expert's knowledge and contributed to his customers his advice as consulting expert. He was senior partner in the firm of Hill, Clarke & Co., and made the economical use of steam power his particular field. In 1873 he was appointed, in connection with C. F. Adams, Jr., a commissioner for the State of Massachusetts to the Vienna Exposition, and published a report upon the exposition in general and a special report on the machinery exhibits. He was later one of the commissioners to organize the Massachusetts department of the Philadelphia Exposition. At Vienna he was vice-president of the Patent Congress there held. He retired from business in 1895, but practised as expert until his health failed. He died in Boston March 18, 1899.

Besides his contribution to the *Transactions* of the Society Mr. Hill had prepared and delivered a considerable number of lectures on scientific subjects.

ALFRED E. HUNT.

The death of Mr. Hunt is as much a casualty of the Spanish-American War of 1898 as any of the losses in the field.

Alfred Epher Hunt was born March 31, 1855, in East Douglass, Mass. He was educated at the Roxbury High School, and graduated in 1876 in the department of metallurgy of the Massachusetts Institute of Technology. For some time after graduation he was connected with survey work, both geological and for railroads, but later he entered the Bay State Iron Works in South Boston as chemist, where the second open-hearth furnace in America was erected. From 1877 to 1879 he was manager and chemist, superintending the steel department

of the Nashua Iron and Steel Company. He then went to Pittsburg as superintendent and chemist with Park Brothers & Co., managing the open-hearth and heavy forging department of the Black Diamond Steel Company. In 1883 he resigned to join Mr. George H. Clapp to form the firm of Hunt & Clapp, establishing the Pittsburg Testing Laboratory, making a specialty of physical and chemical testing of materials and the inspection of iron and steel structures. Important metal structures of the past fifteen years have passed under their hands, such as the Cairo bridge over the Mississippi, the Omaha bridge over the Missouri, the Thames River bridge at New London, and the Poughkeepsie bridge over the Hudson.

In 1888 Captain Hunt organized the Pittsburg Reduction Company to manufacture aluminum according to the process discovered by Charles M. Hall. Previous to this time the metal had sold for $15 a pound, so that it was practically a precious metal, but as produced by the new process it dropped to prices ranging between thirty and forty cents per pound. The successful completion and development of the Niagara Falls works of the same firm gave promise of further production.

Captain Hunt entered the National Guard of Massachusetts at an early age, rising to the position of a commissioned officer. He identified himself on his removal to Pittsburg with the Pennsylvania National Guard, and ultimately rose to the position of Commander of Battery B. He left Pittsburg April 27th for Mount Gretna, and after the necessary service at Chickamauga they left July 28th for Porto Rico, where they landed August 5th, and on August 12th began the march to Guyama. The battery was just ready to fire its first shot in the Cuay Mountains when the news of the protocol was received. Returning, the battery was reorganized, and once more mustered into the Pennsylvania militia. Captain Hunt was taken ill at Chickamauga and was invalided, but on the departure to Porto Rico he determined to accompany it before he was fully recovered. He contracted malaria, which affected the heart action, and though he made a trip in April to the seashore hemorrhages followed an exertion and he passed away April 26, 1899.

Captain Hunt was a member of the party of engineers who went to England in 1889, and by reason of his active connection

with the Institute of Mining Engineers he was their representative in official business of the party. He connected himself with the Society on the return from Europe at the New York meeting of 1889.

R. W. BELLHOUSE.

Mr. R. Wynyard Bellhouse, who was killed by a fall in Detroit, April 3, 1898, was born in Manchester, England, in 1856, and served his engineering apprenticeship at the celebrated locomotive works of Beyer, Peacock & Co., Manchester; then directed the erection of the Eagle Iron Works, of which he later became superintendent and partner in the same concern, building bridges, roofs, etc.

From 1881 till 1883 he was with the Lancashire & Yorkshire Railway, when he came to Syracuse, N. Y., as chief engineer for the Sanderson Brothers Steel Company, largely controlling the reconstruction of the works. In 1886 he was doing general engineering work in Syracuse, building Portland cement works for the Warner's Portland Cement Company, and later similar works in Ohio. At the time of his death he was employed by the Solvay Process Company at Detroit as erecting engineer.

Mr. Bellhouse had by faithfulness to duty built himself up to be a valuable man, and his death was a loss, and deeply regretted by the friends he had made in this country. He joined the Society in 1886.

LEWIS MILLER.

Mr. Lewis Miller was of German stock; his ancestors were pioneers of Pennsylvania, his parents going to Ohio about 1812. They were industrious, frugal, God-fearing people. Lewis Miller was born in Greentown, Stark County, Ohio, August 24, 1829. Until he was seventeen years of age he worked on his father's farm, attending district school in the winter. For the next five years he divided his time between working part of the time, attending Plainfield Academy in Illinois, and teaching school in Stark County. Mr. Miller took up the invention of agricultural machinery and was one of the first persons to produce practical mowing and reaping machines. In 1852 he became partner in a small shop in Canton, Ohio, and engaged in the manufacture of mowers, reapers, and threshing machines.

Mr. Miller soon showed such extraordinary skill and inventive talent that his harvesting machinery, bearing the name of "Buckeye," found its way into every part of the grain-producing world.

In 1863 the business in Canton had grown so large that the company was under the necessity of branching out, and a large shop was built at Akron, Ohio, to which place Mr. Miller moved. He was the active superintendent of this shop from 1863 until the end of his earthly career, February 17, 1899. Among his many inventions the most marked one was made about 1855, and consisted of the double-jointed bar and the use of two driving wheels; this enabled the cutter bar to conform itself to the inequalities of the ground, also enabling the finger bar to fold over the machine, for convenience and safety in moving, when not at work. The method of gearing this machine was a new departure, and is in use, with slight modifications, on all mowers and reapers to-day.

Each succeeding year of his long and busy life was full of new ideas and inventions in mowers, harvesters, binders, corn harvesters, and other machinery, with the ever changing details suggested by experience in the harvest fields of not only our own but of foreign countries.

Mr. Miller became a member of the American Society of Mechanical Engineers in 1883. He never had the advantages of a technical education, but was endowed with inventive talent, fine judgment, great perseverance, endurance, and a fine physique. He had a strong will, coupled with a cheerful hopefulness, that enabled him to keep on in the face of seeming defeat and eventually achieve success. He had good judgment in the selection of his assistants, and by fair treatment secured their best coöperation.

The builder of agricultural machinery has not the opportunity to build monuments in engineering matters like great bridges, engines, steamships, and railways, but taking into account "the greatest good to the greatest number," a man with such splendid achievements as Mr. Lewis Miller is entitled to much credit.

Mr. Miller was a many-sided man, and each side good and strong. He was a Sunday-school superintendent for forty years and one of the leading Sunday-school men of the country. He was active in the improvement of the public schools of his

own city, serving on the school board for thirty years, a large part of the time as president. He had ideas of his own, both on education and buildings, and gave much time and care to putting these into practical use. He was also active, both personally and financially, in the upbuilding and progress of several colleges, particularly Mount Union College. He was a member of the board of trustees thirty-four years, and president of the board for thirty-two years. He was also on the board of trustees of two other colleges.

Mr. Miller was the founder of Chautauqua Assembly and a large contributor to it. Chautauqua was his recreation, and he worked out the details, planning walks, drives, buildings, and amphitheatre, as well as other features which have made Chautauqua so attractive to thousands. The main point, however, with him was the educational and Christian uplift of the people.

He died February 17, 1899, as the result of a dangerous surgical operation.

MAURICE A. MULLER.

Mr. Muller was born at Berlin, Germany. After an apprenticeship and three years at the Polytechnic he became a junior draftsman, after which time he went to England. He there served five years as a designer in the locomotive and bridge department of the Great Eastern Railway Company, serving thereafter for four years as chief assistant to the late Robert Mallet of London. During the Vienna Exhibition in 1873 he represented *Engineering*, of London, as expert correspondent. The next succeeding years were passed in Germany, France, and England, partly in superintendence of the erection and operation of English machinery, and partly with literary work. He then came to the United States, and was for many years chief draftsman at the Weston factory of the United States Electric Lighting Company, of Newark, which was his position when he connected himself with the Society at the Boston meeting of 1885. In the last part of his life he was engaged in business upon his own account, in literary work, and in service as professor of mechanical engineering at the Newark Technical School. He died May 21, 1899, being at that time superintending the construction of the new Weston factory. His death resulted from an exposure at the works, which brought on a cold from which he never recovered.

CHARLES A. BAUER.

Mr. Bauer was one of the charter members of the Society, his application letter being dated May 14, 1880. He began his professional work as apprentice at the gun and lock smithing business, and the end of the war found him engaged as machinist and student of mechanics at the shops of Miles, Greenwood & Co., of Cincinnati. In 1867 he became teacher of drawing and mechanics in the Ohio Mechanics' Institute of that city. In 1871 he moved to Hamilton, Ohio, to become general superintendent of the Niles Tool Works, from which he was again promoted to become consulting engineer with Lane & Bodley of Cincinnati. In 1875 he began his best-known work in agricultural machinery as superintendent of the Champion Bar and Knife Company, of Springfield, Ohio. He was in this position when he connected himself with the Society.

Like most of the veterans, his experience covered a wide range of subjects, "including most prominently the design and practical execution of nearly all kinds of machine tools, stationary and river engines, rolling mills, furnaces, pumping, mining, and general milling machinery." He further stated in his application that he had also given some practical attention to the establishment of plants and the arrangement and details of buildings on grounds for manufacturing purposes.

He was elected a manager of the Society in 1894, and died June 12, 1899.

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